AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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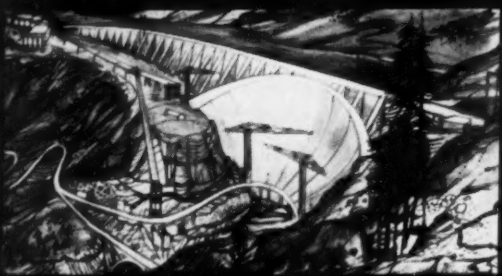


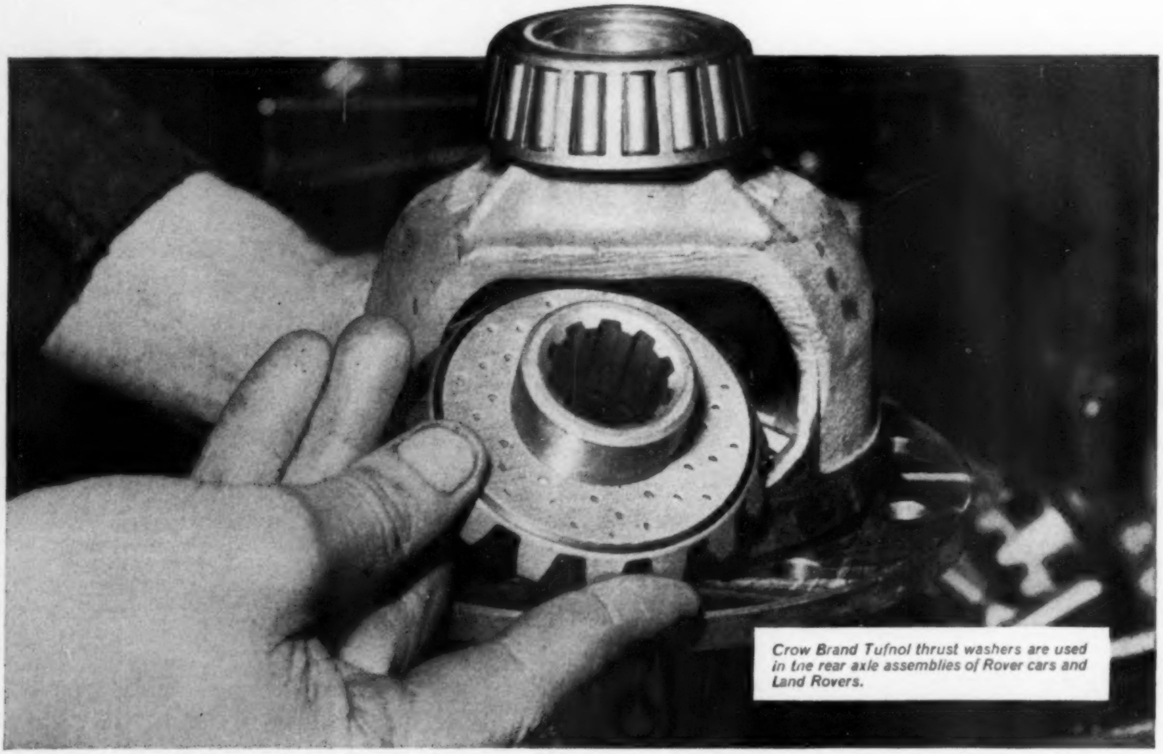
MORE POWER FOR FRANCE
To Electricité de France the Savoy Alps mean power. The Roselend Valley project, for example, involves building two dams and a 476,000kW power station—the most powerful in the country—which will increase France's power reserves by 1,000 million kW a year. Atlas Copco equipment has been used almost everywhere at Roselend—the main tunnels being drilled with more than 100 Atlas Copco rock drills and many thousands of Sandvik Coromant steels.

THROUGH THE HEART OF MONT BLANC
On completion of the seven-mile Mont Blanc road tunnel motorists will find Paris 137 miles nearer Turin. For the first time they will be provided with an Alpine link between France and Italy open all the year round. More than half the tunnel is being driven from the Italian side using Atlas Copco rock drills fitted with Sandvik Coromant drill steels and powered by Atlas Copco compressors.

OIL PROSPECTING IN THE SAHARA
In the continuing search for hidden oil resources in the Sahara, seismic prospecting techniques are often employed. Many prospecting companies use Atlas Copco equipment to drill the hundred or more sounding holes for each blasting pattern. Typical of the units used for this work is the Atlas Copco truck-mounted compressor—air cooled for desert operations—driving a chain-fed rock drill with Sandvik Coromant drill steels.

Atlas Copco's contribution to the jet age—an Air Partner starting a jet airliner at London Airport

Have you discovered TUFNOL?



Crow Brand Tufnol thrust washers are used in the rear axle assemblies of Rover cars and Land Rovers.

Tufnol is an extremely versatile material with many potential applications in the automobile industry. It is light but strong and hardwearing, resistant to corrosion and requires little or no lubrication. Tufnol is an excellent electrical insulator, it machines easily with ordinary tools and resists deterioration in storage. In short, Tufnol is a single material, incorporating the virtues of many.

These virtues have already been recognised by Rover, Jaguar and Burman & Sons, amongst others, in the selection of Tufnol for small components giving vital service. The Rover Co., for instance, use Tufnol thrust washers in the rear axle assemblies of their cars and Land Rovers. Jaguar cars have Tufnol washers and bushes in the gearbox and linkage, as well as washers between clutch and brake pedals. And Burman & Sons have this to say about their Tufnol steering column bushes: "Tufnol gives a good bearing for the situation which has no positive lubrication; the bushes are light and they do not corrode".

There are twelve brands of Tufnol—all are laminated plastics. Each brand of Tufnol has pre-determined properties, formulated to satisfy specific requirements, and each is available in sheets, rods, tubes, angles and channels.

Your local Tufnol Branch Office will gladly provide you with further information and technical advice.

'CROW BRAND'

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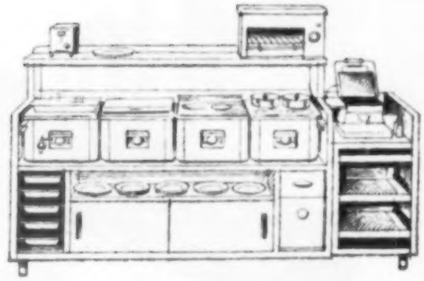
424

Canteen Catering



As in every other industry, the main factors governing the economics of the catering business are: quality of product, cost of production, and well-being of workpeople. Electricity measures up well to these three essential factors.

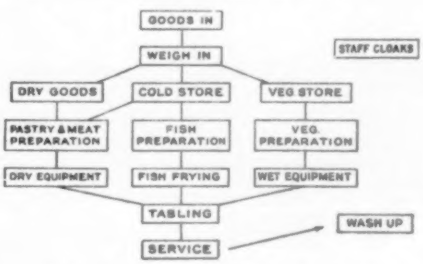
The outstanding virtue of electricity is better cooking, particularly in roasting and pastry ovens where high quality is more easily maintained than in ovens using other forms of heat. The cost of production varies somewhat with the type of food and the size of the establishment, but is usually between $\frac{1}{2}$ and $\frac{3}{4}$ of a unit of electricity per meal. Cleanliness of electric cooking is axiomatic and provides better working conditions for the staff.

The actual size of the kitchen is influenced greatly by its shape and by the number of people catered for, but a rough guide is as follows:

KITCHEN TO DEAL WITH	SIZE
up to 100 persons	5-6 sq. ft. per person
100-250 persons	4-5 sq. ft. per person
250-1000 persons	3-4 sq. ft. per person
over 1000 persons	3 sq. ft. per person

Design

Where the kitchen is designed from the start for the full use of electricity, planning is simplified as the equipment can be placed where it is required without reference to the need for flues.



If an existing kitchen is already using other types of cooking equipment, however, electrical equipment can still be introduced item by item to bring increasing benefits.

Quick-service Equipment

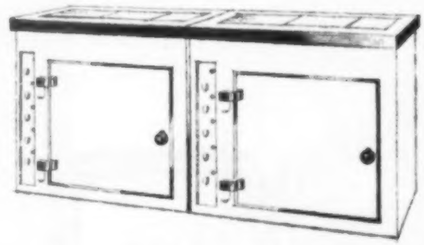
The popularity of the quick-service establishment where the food is cooked at the service counter in the full view of the customer is steadily growing, and this type of catering can readily be provided in the canteen by the installation of a Back Bar cooking unit, installed behind a section of the service counter.

The popularity of the mid-day joint and two vegetables is on the wane and the really up-to-date canteen should provide the welcome alternative of fresh food cooked on the spot.

Electric Catering Equipment

Electric catering equipment covers every single kitchen activity and some of the appliances in common use are:

COOKING. Ranges, boiling tables, steaming, roasting and pastry ovens, vegetable boilers, fryers, griller/toasters.



SERVICE AND WASHING-UP EQUIPMENT. Bains-marie, hot cupboards, tea and coffee machines, washing-up machines for the larger kitchen and sterilising sinks for the smaller, refrigerated cold-service counter and display cabinets, soda fountains.

PREPARATION. Mixing machines with attachments for chopping and mincing etc., potato peeler, slicing machine.

QUICK-SERVICE EQUIPMENT. Infra-red (contact) grill, automatic toaster, griddle plate, automatic fryer, boiling plates, soup heaters, etc., and, of course, the indispensable refrigerator.

For further information, get in touch with your Electricity Board or write direct to the Electrical Development Association, 2 Savoy Hill, London, W.C.2.

Excellent reference books on the industrial and commercial uses of electricity are available—" Electric Commercial Catering Handbook " (5/-, or 5/6 post free) is an example.

E.D.A. also have available on free loan in the United Kingdom a series of films on the industrial uses of electricity, including commercial catering. Ask for a catalogue.

12799

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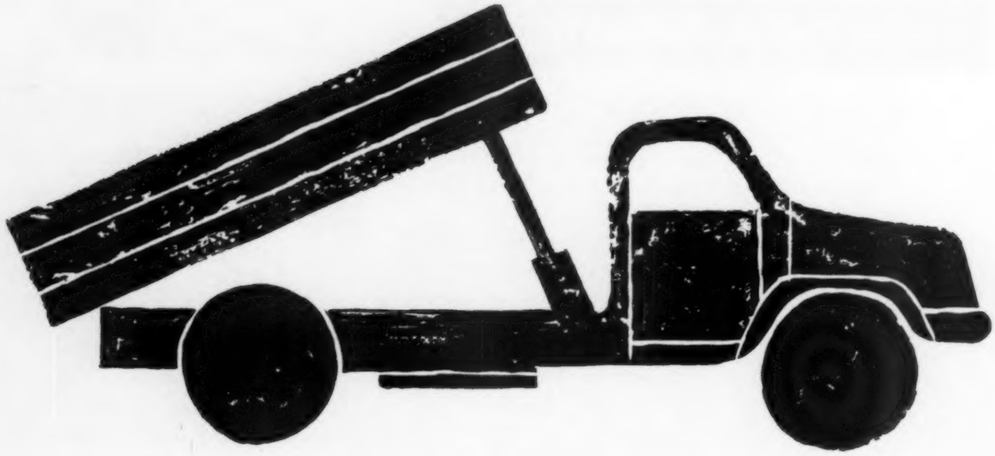

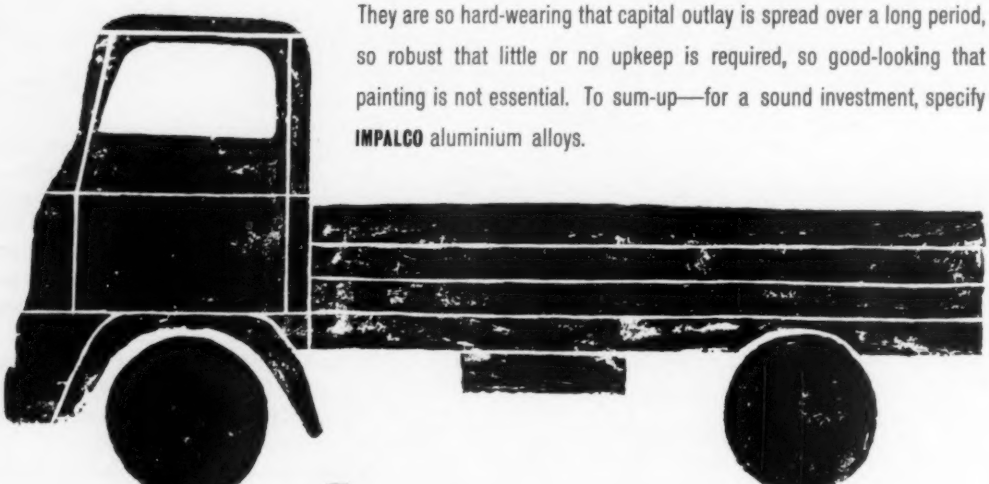

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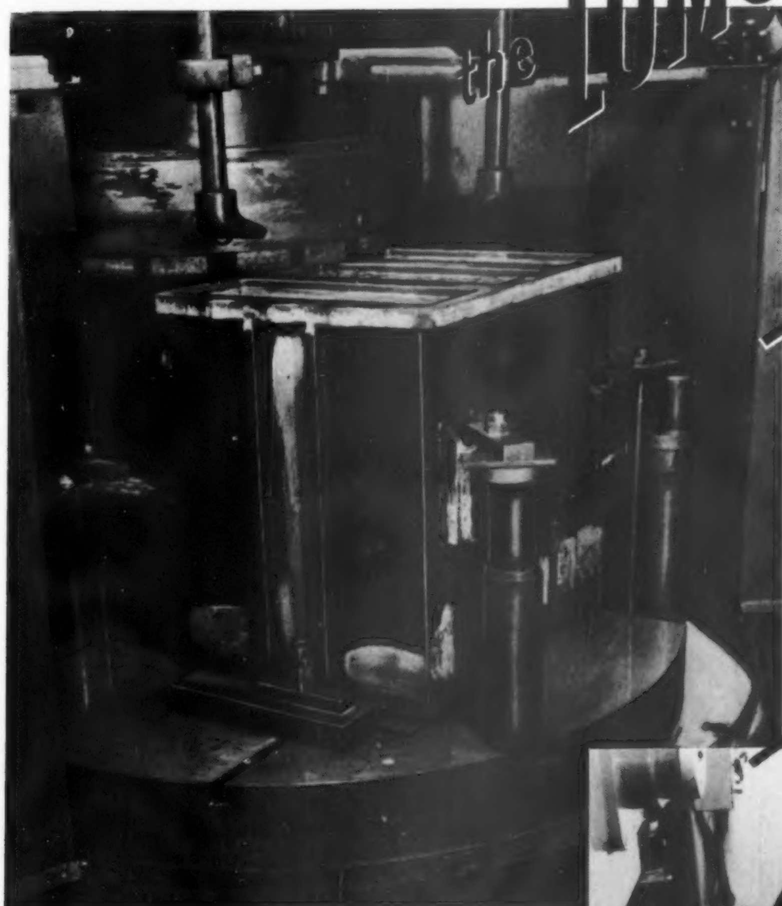
LUMSDEN technique



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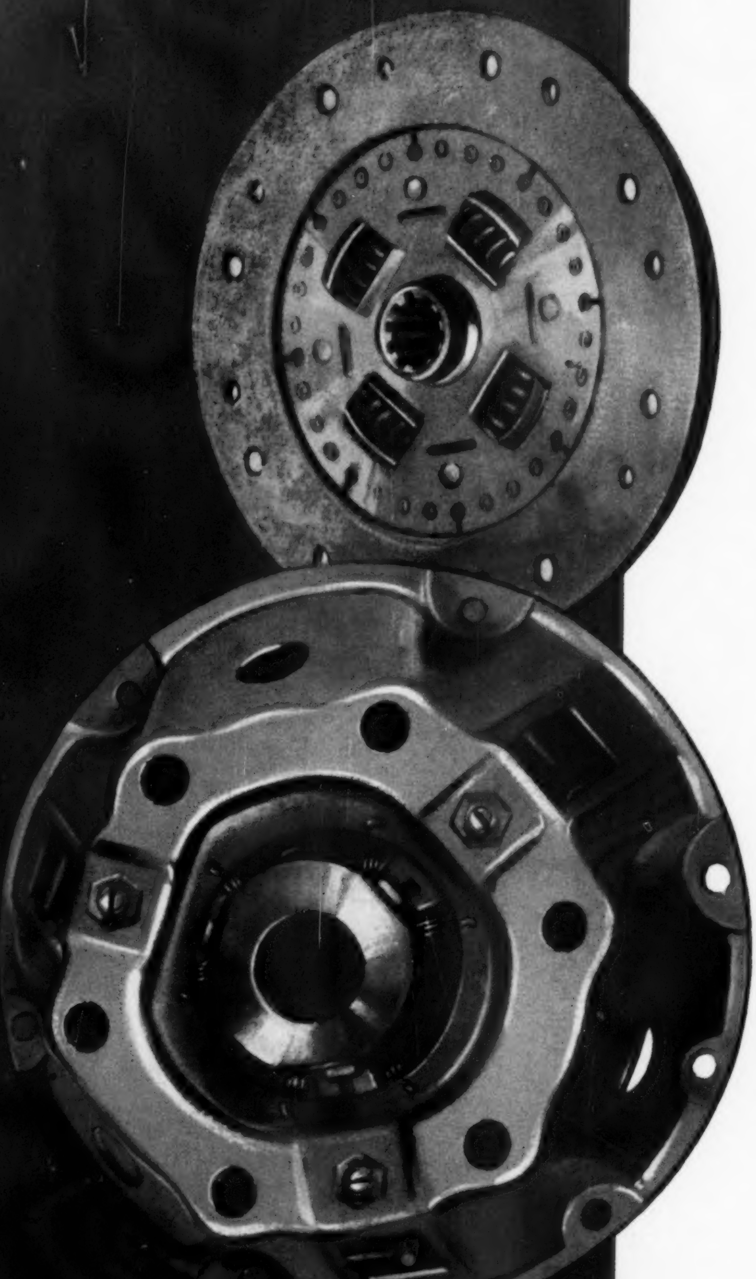

Borg & Beck clutches have a
by a combination of brillia
excellent workmanship.

Basic features include unifor
pressure-plate of great rigidi
release-lever system having t
centre cushion drive plates wi
a great deal to smooth the tra

The illustration at left shows
these are made in the foll
10″, 11″.

The strap drive, shown in the
clutch, permits the pressure-
to-metal sliding contact. It wa
clutches and its success has
extended range is now as fol
and larger sizes on applicatio

In addition, the 18″ R.4 hea
single or twin-plate type.

For tractors and other equip
Borg & Beck clutches, enab
for both propulsion and drivi

In addition there is a wide ra
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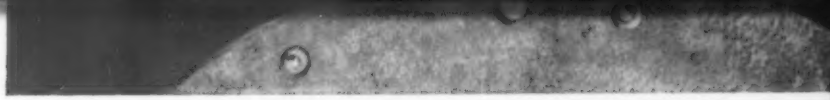
Clutches by BORG

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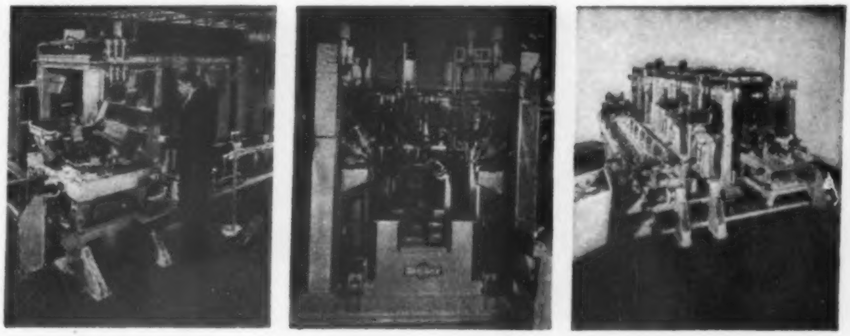
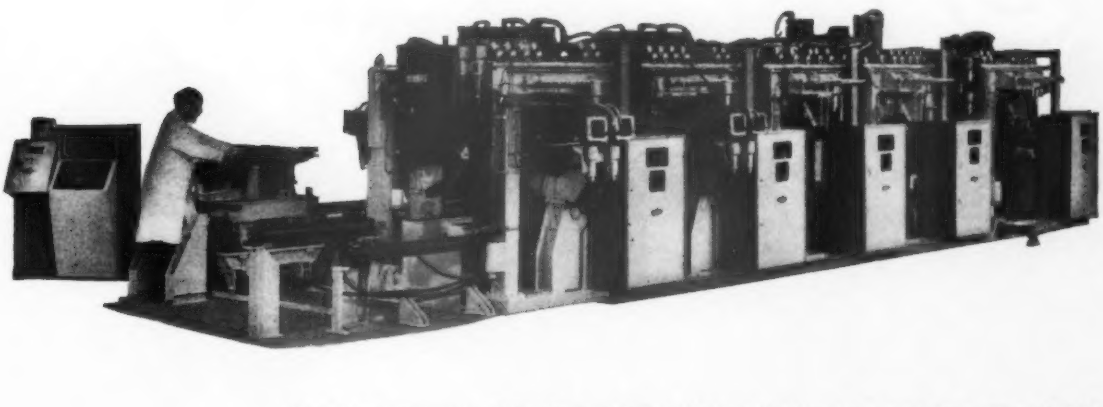
Sheet steel from the City of Steel is also made to a purpose. Each order is treated as a separate assignment so that you, the manufacturer, get the exact kind of steel your product needs. Steel, in fact, made precisely to your particular measure.

Illustrated is a transfer resistance welding machine, recently designed and constructed by Sciaky. It welds the front suspension and engine mounting units of the BMC mini cars in 5 stages. Welding cycle and transfer between stages is automatic.



a welding machine...
...a production line



Every day thousands of Sciaky Resistance Welding Machines are used to produce Britain's automobiles. Working closely with the motor industry, Sciaky has developed a range of welding machines to meet their critical requirements. But, the story does not end there . . . the adaptation of resistance welding to a specific application—the Sciaky technique—is yours for the asking. Let Sciaky experience and know-how work for you, because Sciaky resistance welding techniques are developed to meet your individual needs—whether portable, pedestal, or complete production line.

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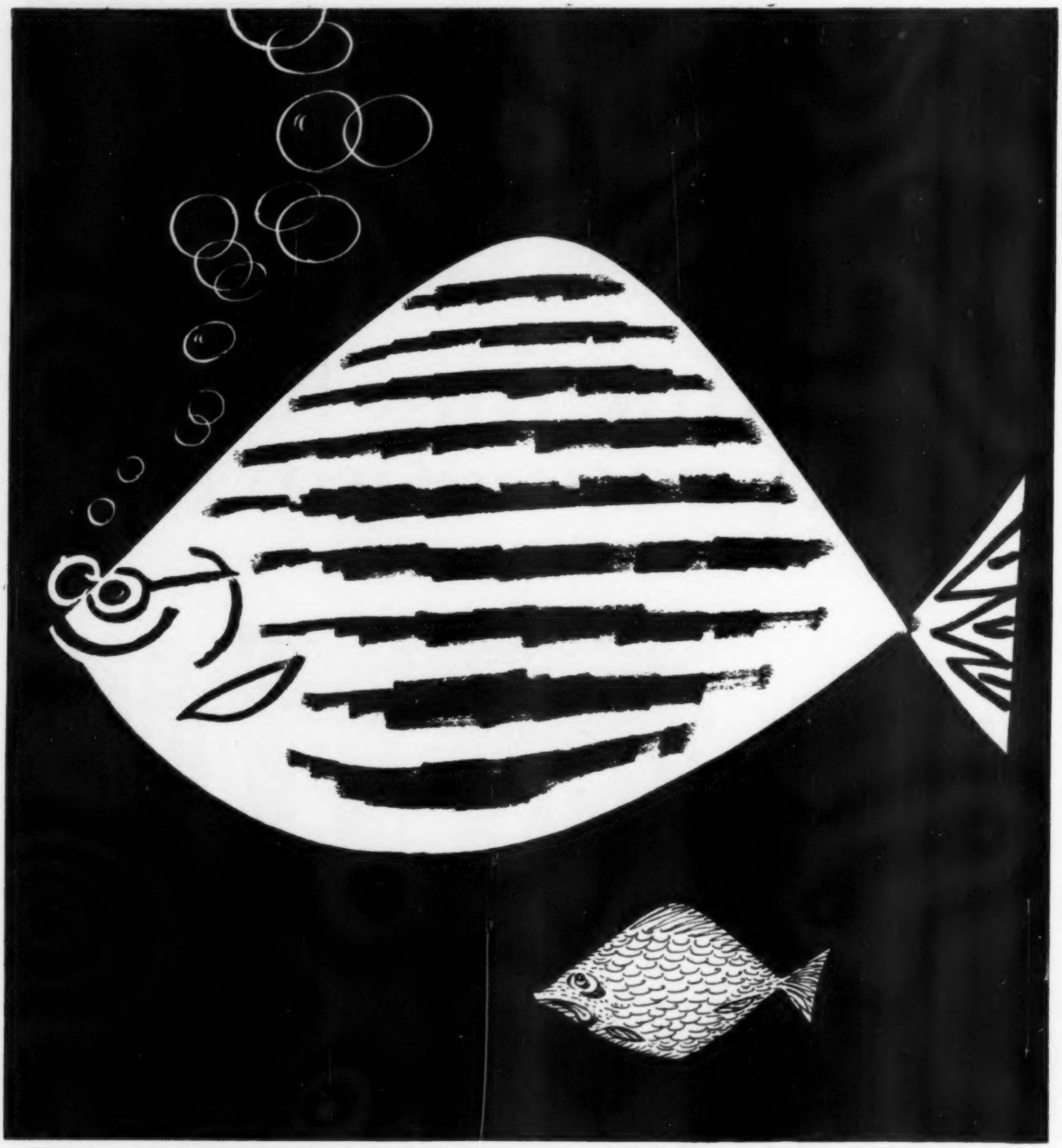

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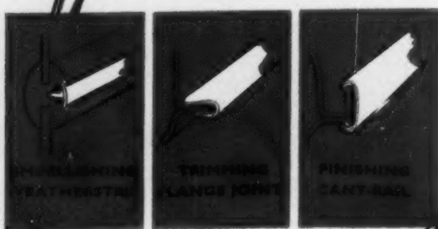

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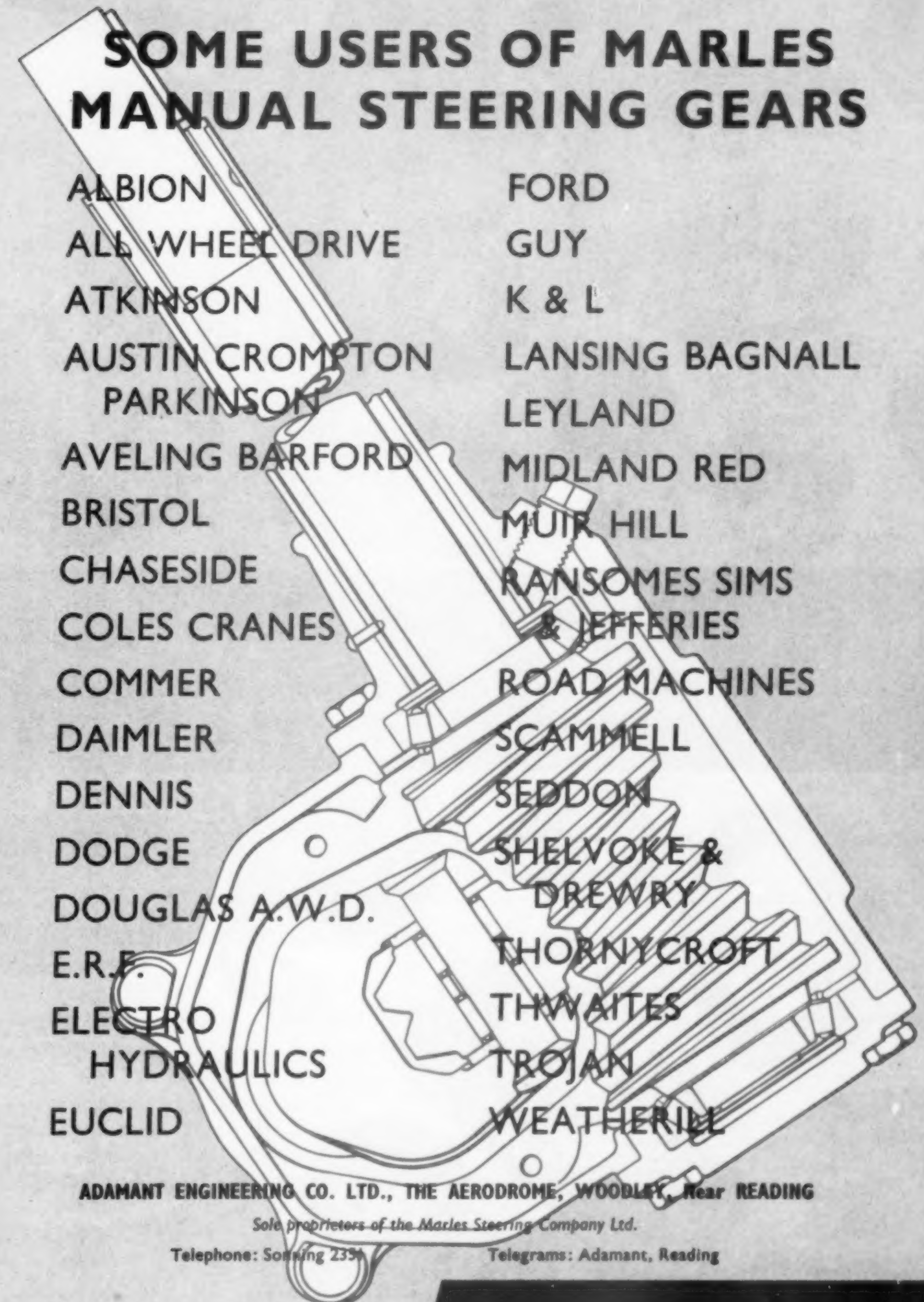

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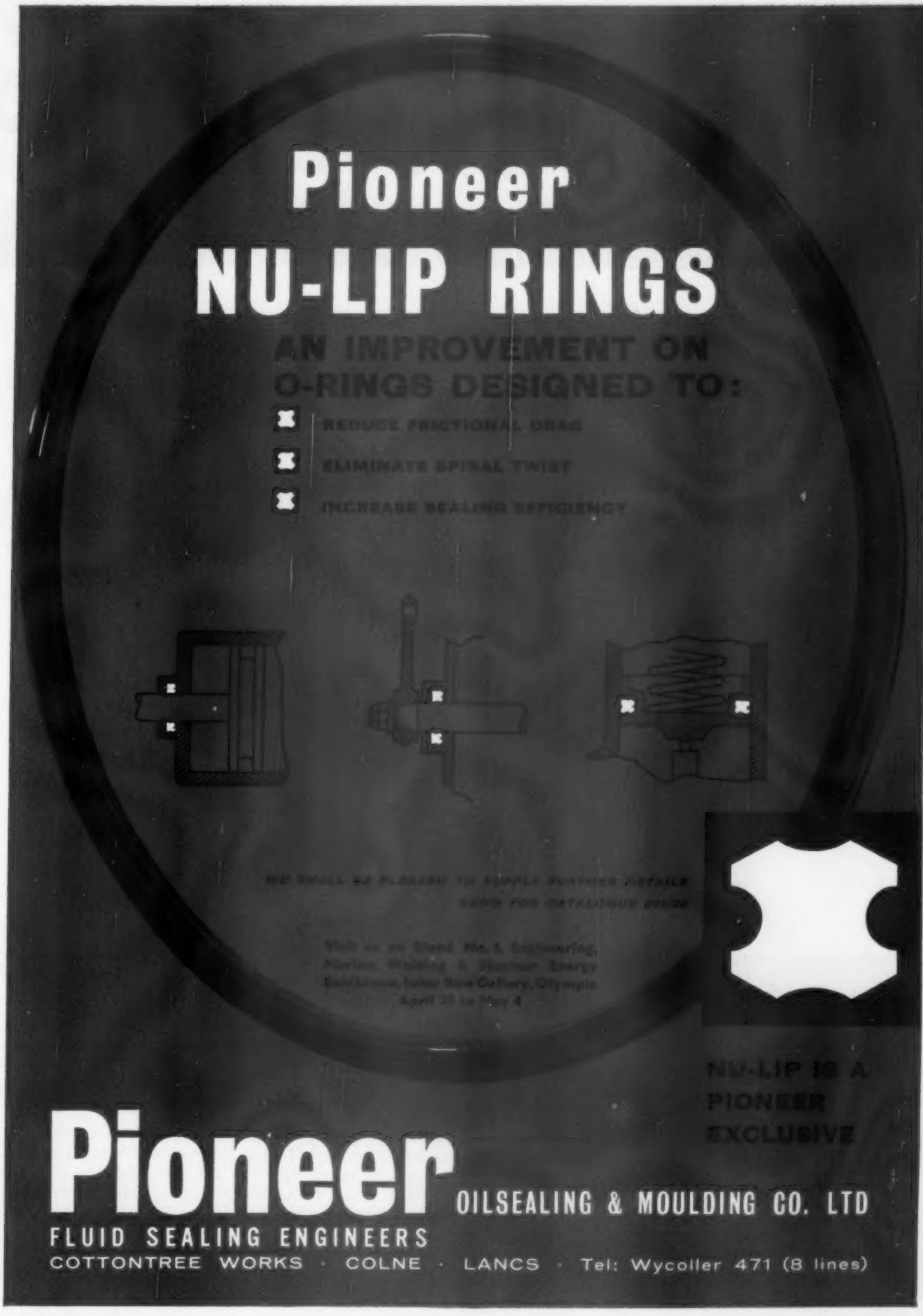
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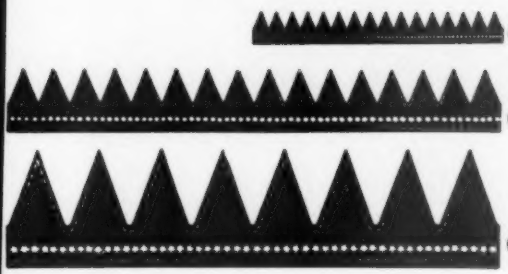

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SIZE 24

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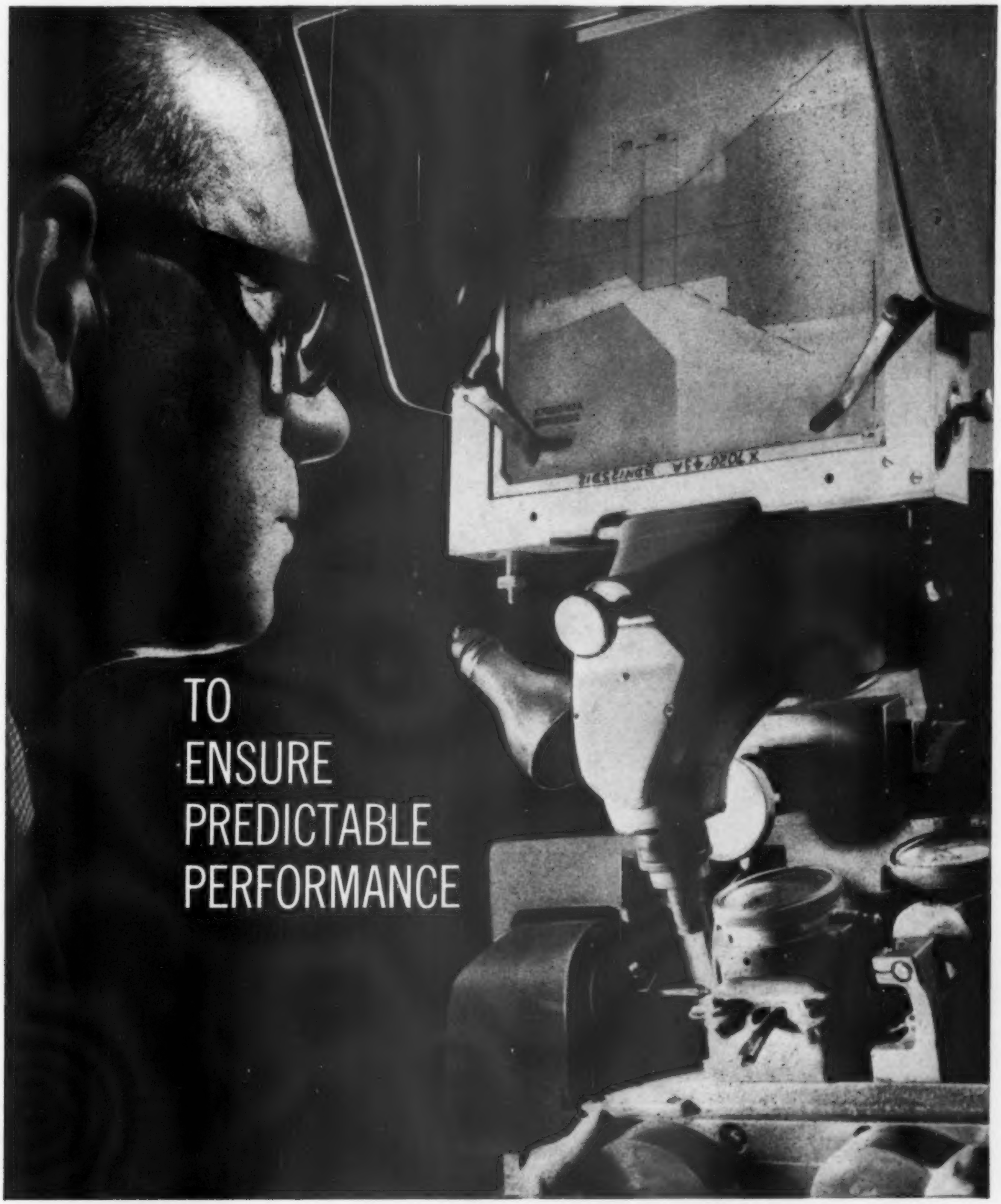
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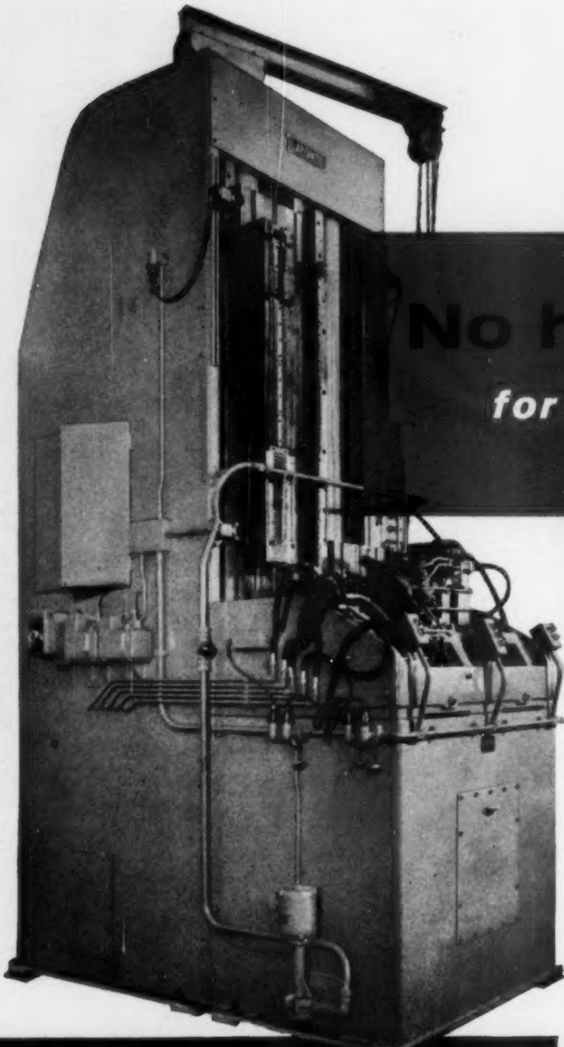
ENSURE
PREDICTABLE
PERFORMANCE



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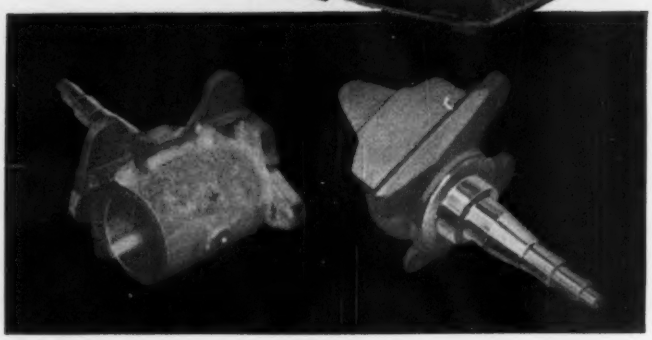
Lapointe design skill does not stop at the basic broaching machine. It goes right through to the design of fittings and fixtures ... to hold all components no matter their shape. The Lapointe 66″ DRV, for example, is now busily broaching Stub Axles. Even these intricately shaped components, with difficult-to-get-at faces, set no problem at all to Lapointe engineers. They adapted the machine, designed the fixture and broach hoist to simplify removal of broach bodies when blades have to be changed. They showed that their ingenuity can surmount all obstacles ... that it pays to come to Lapointe for better broaching.



photographs by courtesy of Armstrong Patents Co. Ltd.

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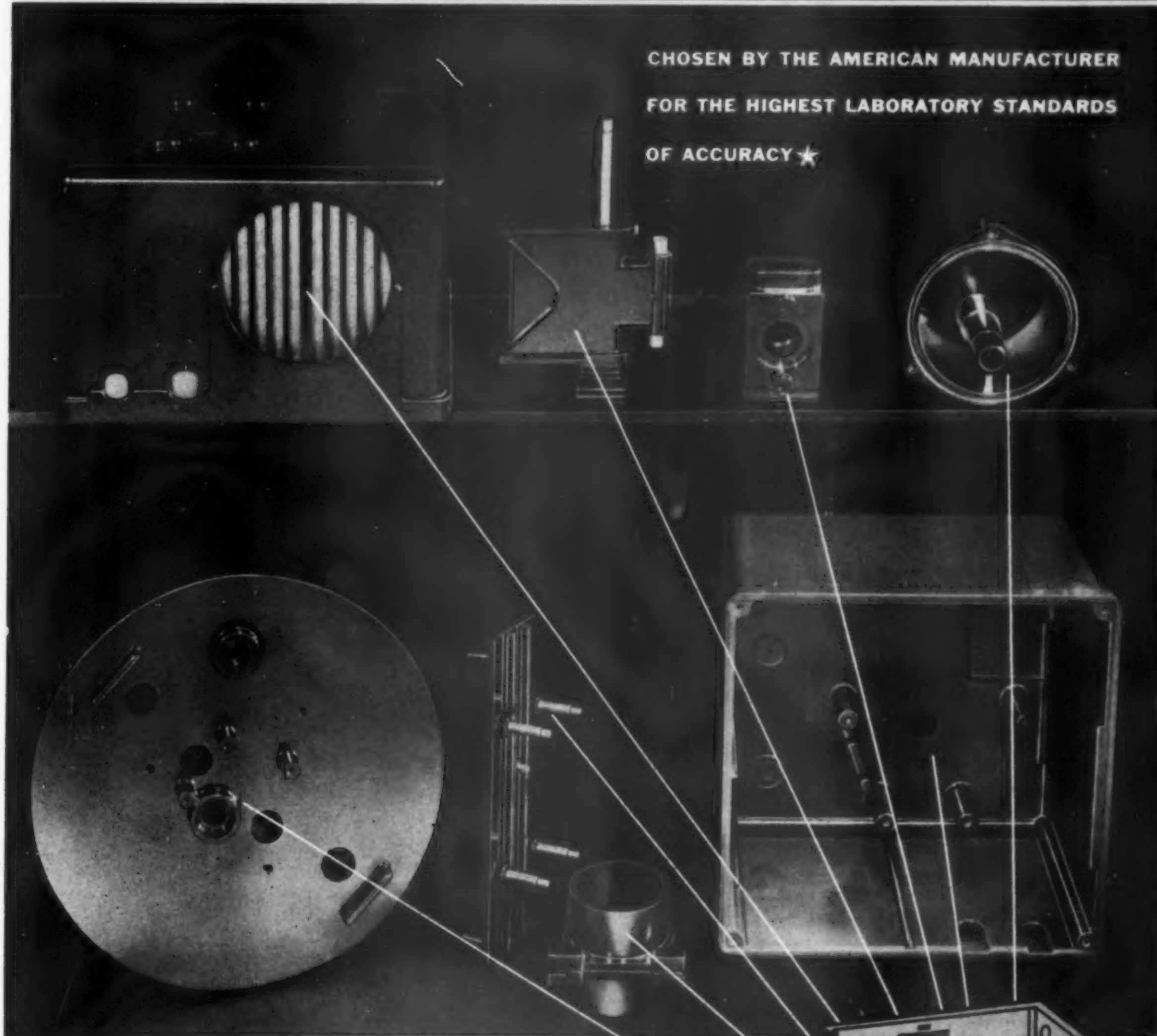

THE TECHNICON AUTO-ANALYZER, a robot chemist, continuously performs everything a chemist does in making a chemical analysis. As meticulous accuracy is required, the TECHNICON INSTRUMENTS CORPORATION of New York chose METROPOLITAN PLASTICS LIMITED to supply the moulds and mouldings in the face of intense British and American competition.

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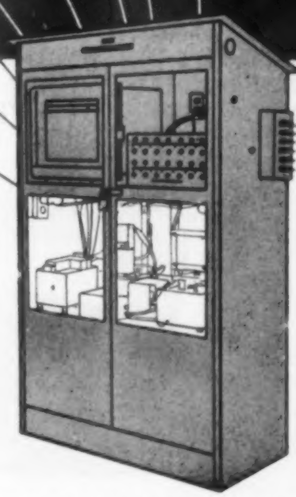
THE TECHNICON AUTO-ANALYZER

This illustration shows a typical Auto-Analyzer as installed in many American industrial plants, hospitals, etc.



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DU pivot bushes for control pedals prevent "stick-slip". The timing chain tensioner sprocket and the gear box remote control incorporate DU.

JAGUAR
Fitted with DU pivot bearings in the clutch and brake pendant pedals.

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Fitted with Borg Warner overdrive, incorporating DU bushes.

HUMBER SUPER SNIPE
DU thrust washers and bushes in the king pin assemblies have caused a marked reduction in steering effort. A DU bush is also used in the gear change linkage.

- Some examples of the extensive use of Glacier dry bearings in private vehicles.

GLACIER *DU BEARINGS

DU dry bearings and bushes are low in cost and are proving their worth on an increasing number of vehicles. Many leading companies in the Motor Industry use them in suspension, steering mechanisms, gear change linkages, brake-clutch controls, etc. They do not require lubrication, have a high load carrying capacity with low friction and "stick slip" is eliminated. Glacier DU accommodates abrasive particles and gives considerably reduced shaft wear.

*Glacier DU dry bearings and bushes are composed of a steel strip lined with a sintered porous bronze layer impregnated with a mixture of a fluoro-carbon plastic (P.T.F.E.) and lead

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Rotella Multigrade flows freely at low temperatures—resists sludge formation. Shell Rotella means immediate cold start *plus* all the proved benefits of Rotella fine quality multigrade oils:

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Measuring the internal temperature of the rubber of an anti-vibration mounting, during dynamic testing.

If you make a product which is subject to vibration, Phoenix can help you!

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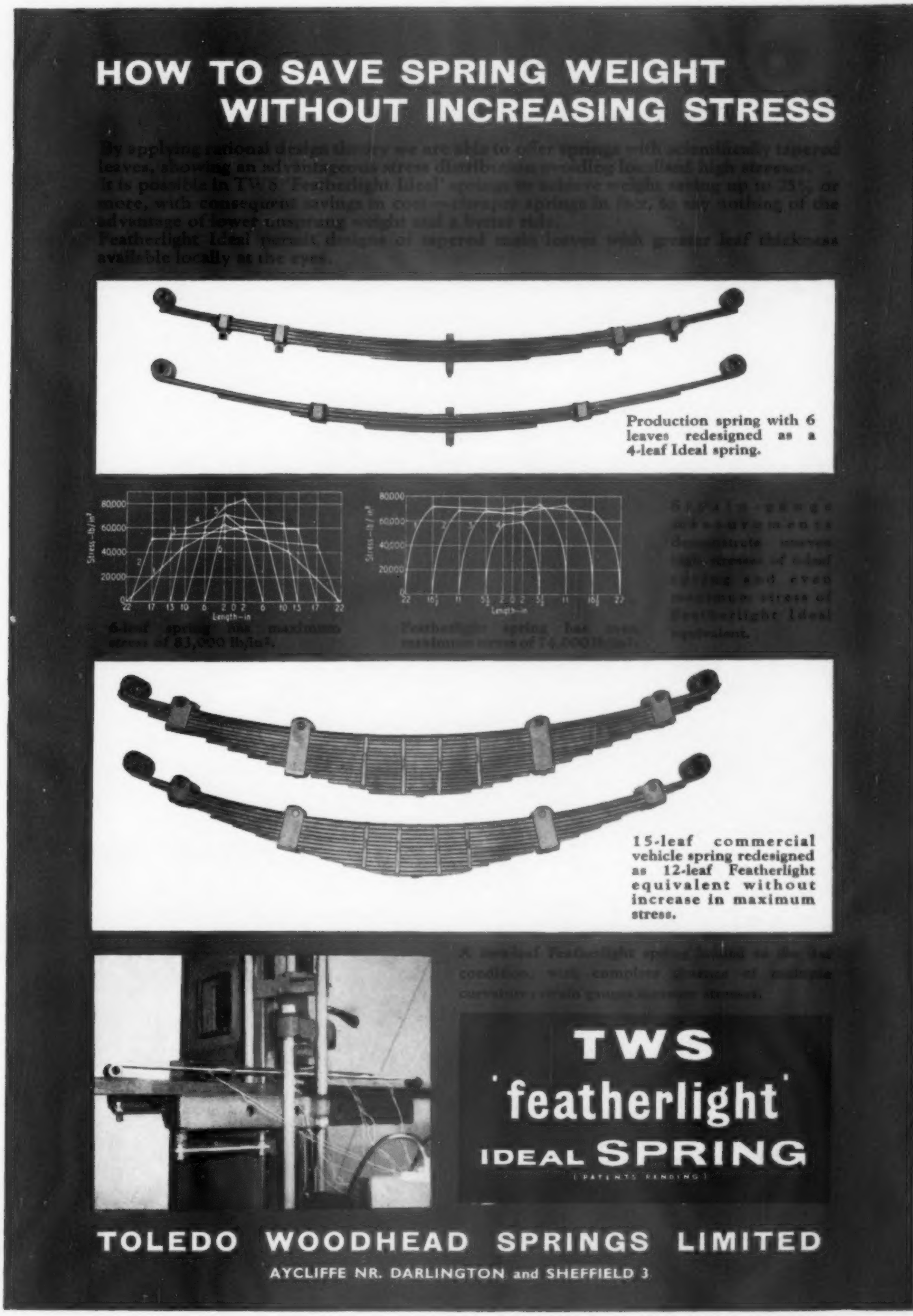
Phoenix Rubber Co. Ltd. Slough, Buckinghamshire. Telephone: Slough 22307/9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WITHOUT INCREASING STRESS
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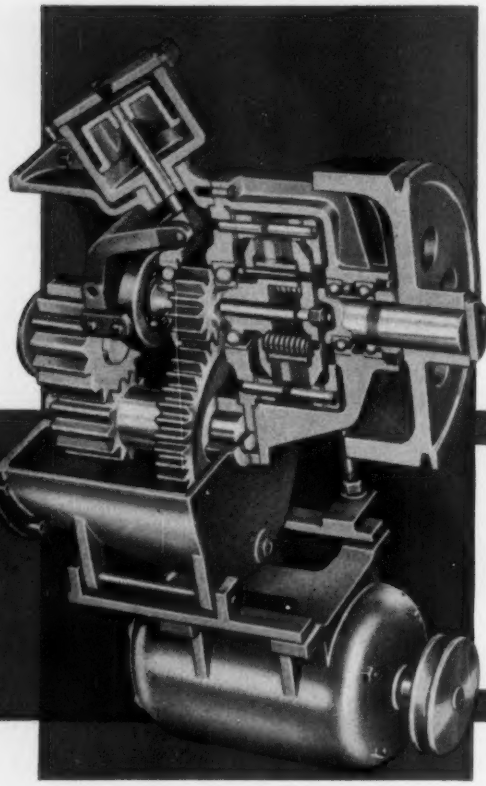

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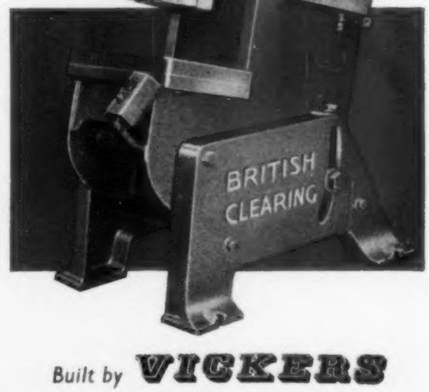

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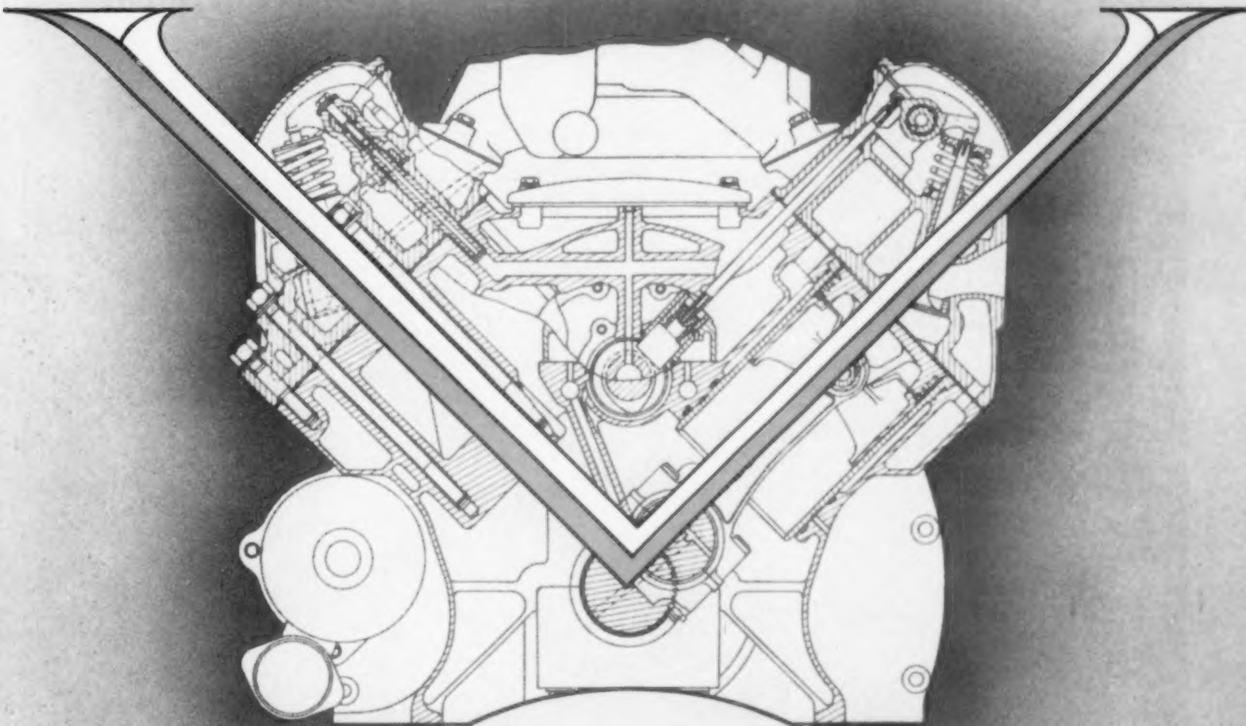

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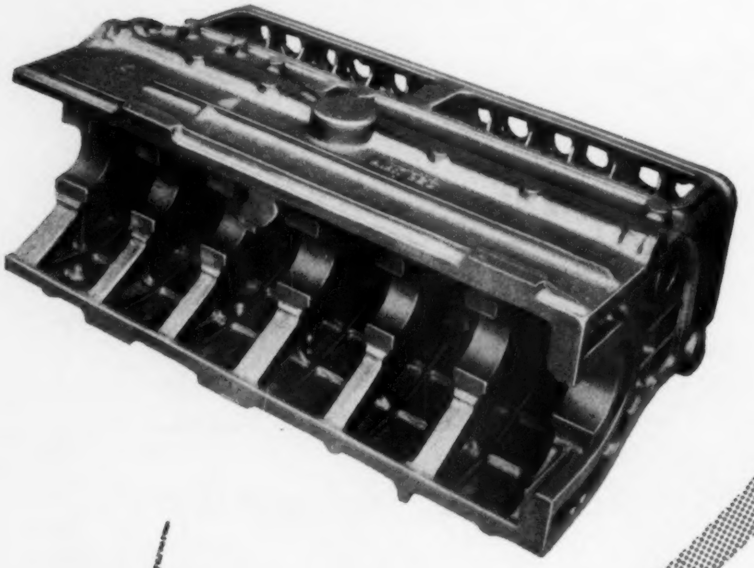

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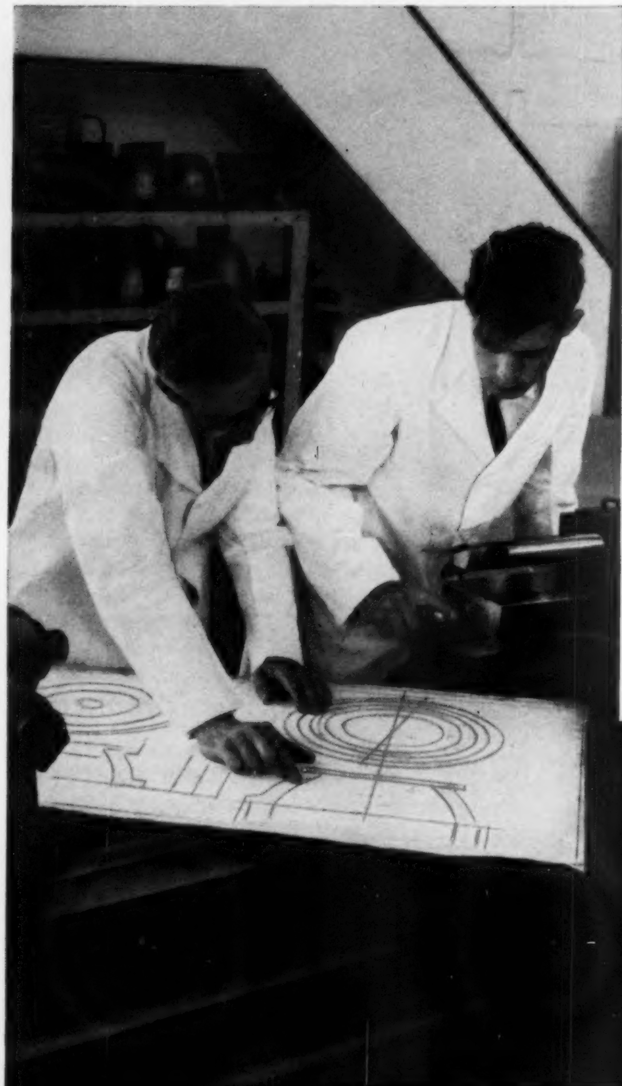

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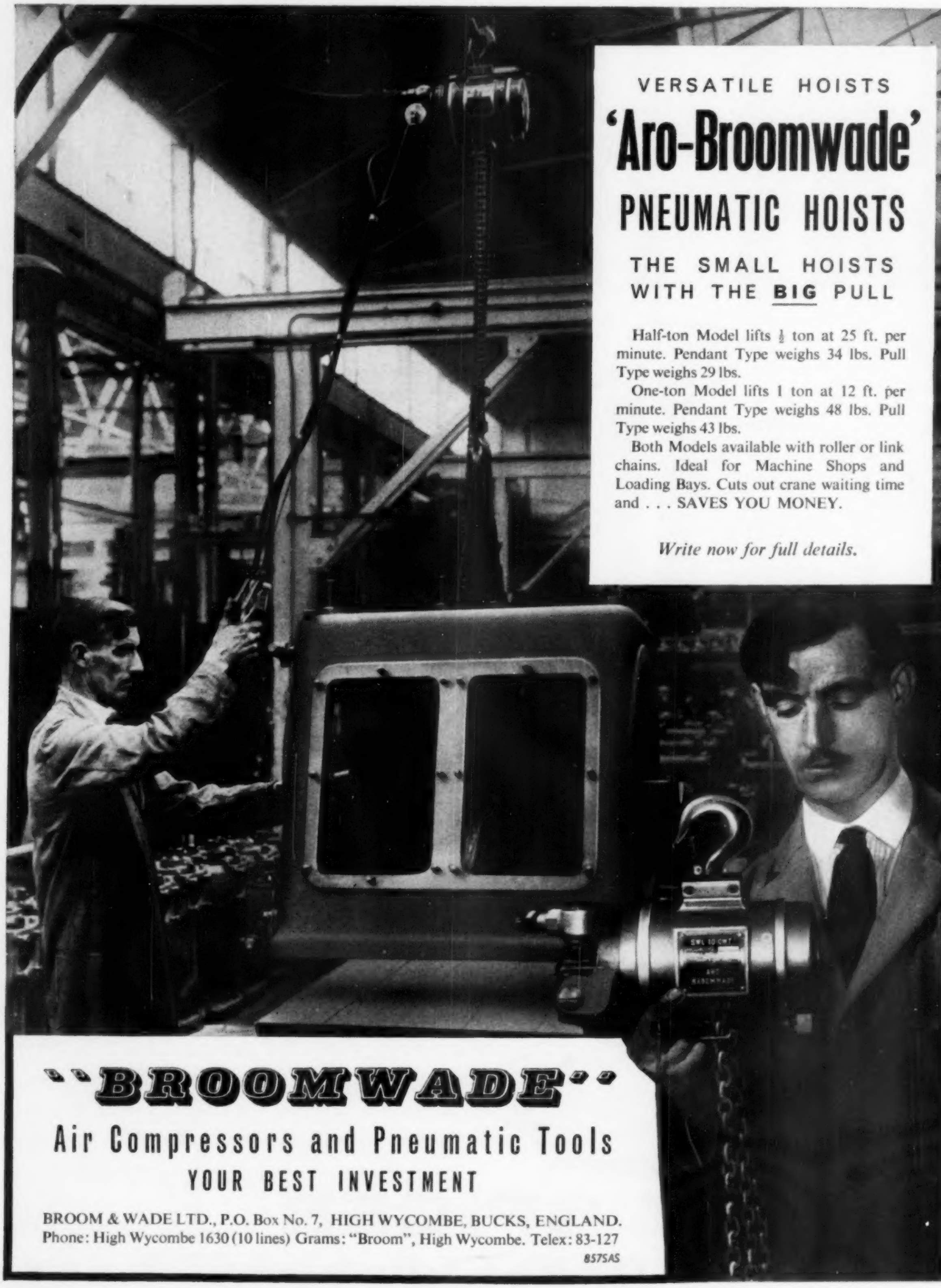

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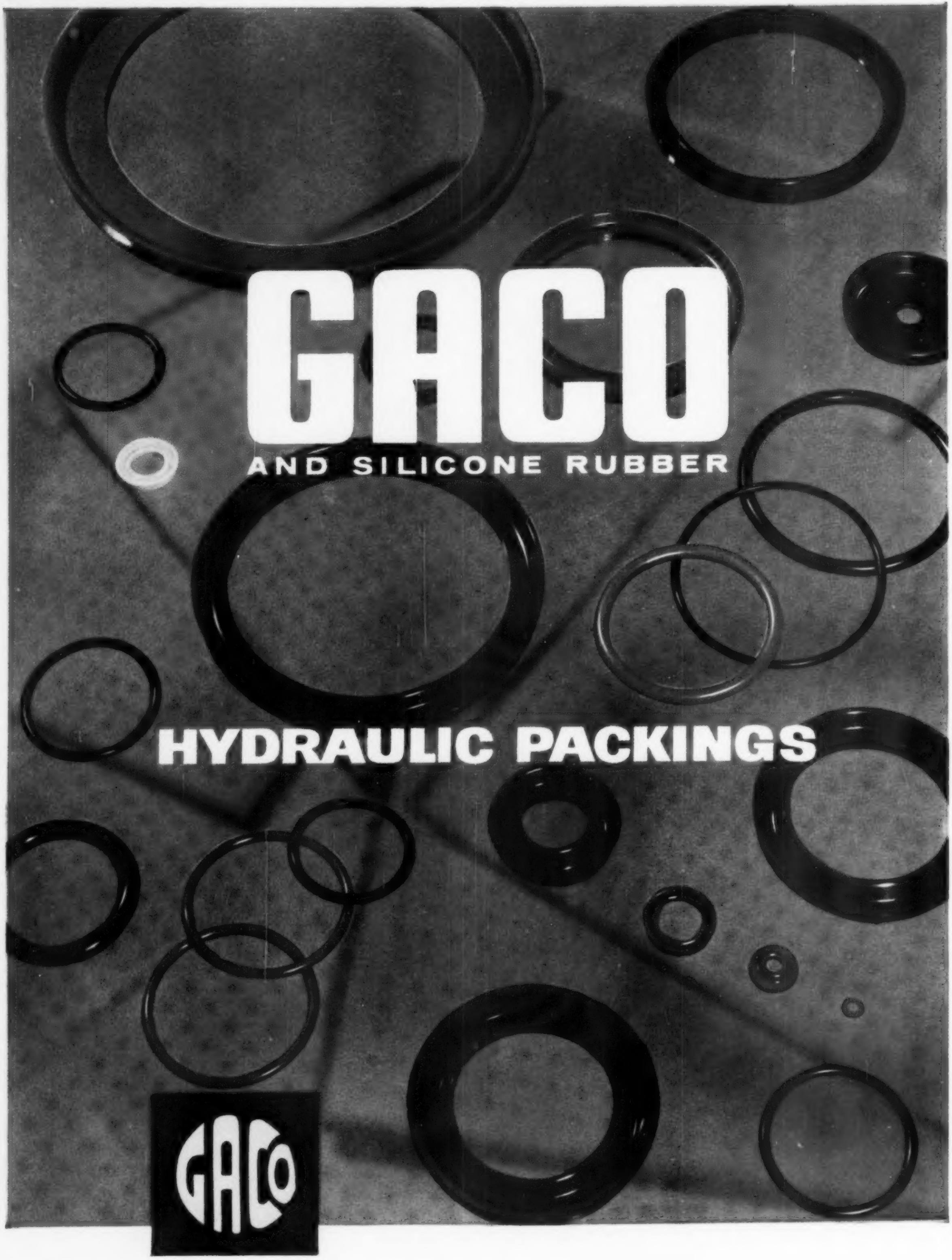
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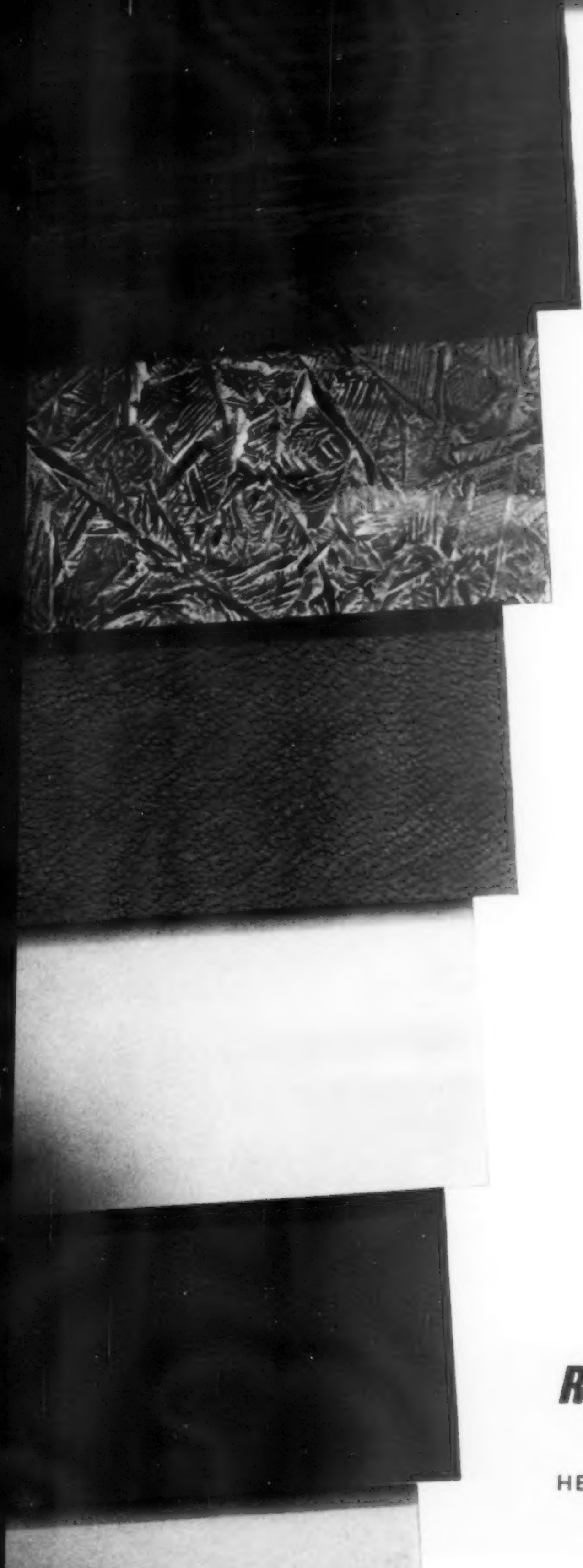

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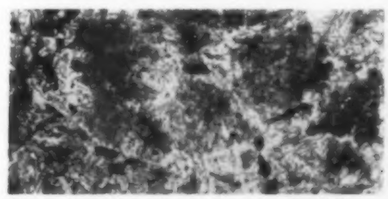
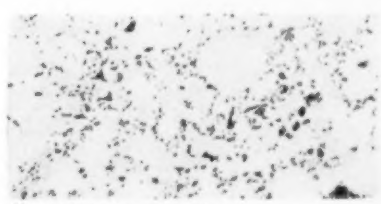


Micro photo shows basic structure of metal



Micro photo shows porosity

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AUTOMOBILE ENGINEER

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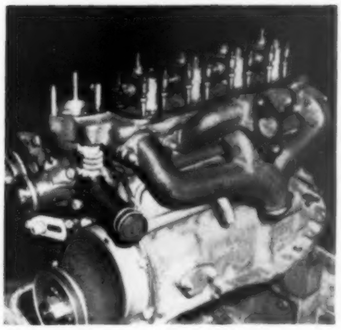

AMONG THE NOTEWORTHY FEATURES OF THE 848 CM³ ENGINE OF THE B.M.C. ADO 15 CAR ARE THE USE OF PRESSED STEEL VALVE ROCKERS AND A SINGLE CASTING FOR THE INLET AND EXHAUST MANIFOLDS

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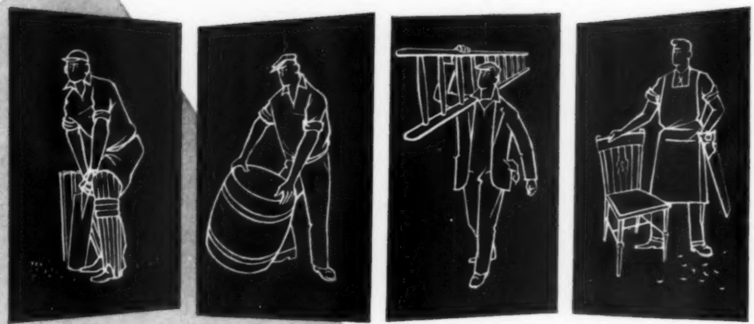

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AUTOMOBILE ENGINEER

DESIGN · MATERIALS · PRODUCTION METHODS · WORKS METHODS

Tank Filler Arrangements

RECENT developments in petrol station forecourt equipment have tended to show up the inadequacy of fuel tank filler arrangements on vehicles. Although the latest equipment available for garages and filling stations is capable of delivery at about 12 gal/min, vehicle filler necks and tanks in some instances will accept the fuel at little more than half this rate.

Blow-backs, often experienced when the modern refuelling pumps are used—especially as the level in the tank approaches to within a gallon or so of the full capacity—cause users of the vehicles to feel aggrieved because of not only the loss of fuel but also the staining of the paintwork. There is another aspect which, perhaps, is not so obvious: it is that site attendants, being unsure as to which vehicles will fill without blow-back, tend to restrict to an unnecessarily low rate the delivery to all of them, and this nullifies the advantages of modern pumping equipment, with the result that there frequently are queues of cars at the pumps and the petrol stations concerned lose customers.

Obviously there is a case for consultation between vehicle manufacturers and the oil industry. The main problem facing automobile designers, of course, is that of making provision for the air to be displaced from the tank during the whole of the time that the fuel is being delivered into it. This is complicated by the fact that the liquid entering tends to drag air in with it, or at least to impede the outward flow, and emulsification can take place in the filler neck, especially if not straight.

One solution is the fitting of a vent pipe of a diameter large enough to allow the displaced air to pass out through it instead of up the filler tube. A simpler expedient is the employment of a filler tube of adequate diameter and optimum shape and cross section: experiment tends to show, for example, that an elliptical cross section would be better than a circular one. Ideally, as regards preventing blow-back, the junction between the lower end of the tube and the tank should be at the centre of the top panel of the tank. The position of the upper orifice of the tube is also important in respect of the avoidance of damage to paintwork during refuelling operations. Since vehicles may be refuelled from either side, the ideal position for a single filler is midway between the sides of the rear skirt panel; although this is not always practicable, the placing of the filler orifice on one side, forward of the rear wheels, surely could be avoided.

In discussions on this subject, it is necessary to avoid thinking solely in terms of the improvement of current practice. As standards of living increase and labour costs become still higher, a point will be reached when no filling station proprietor will be able to bear the cost of operating two or three shifts of pump attendants for seven days a week. This inevitably will lead to strong demands for the provision of slot machine type self-service equipment. Although in many countries legal requirements cannot currently be met by such arrangements, they may be changed when the need becomes more pressing.

The main problem, that of eliminating fire risk, might be solved by fitting a device such as a conical, flexible diaphragm over the tank filler aperture and pushing the delivery nozzle through a central hole in the diaphragm —the edges of the hole would, of course, have to seal effectively round the nozzle, and venting would have to be studied. It would be relatively easy for a pressure actuated switch to be incorporated, as a safety measure, to stop delivery as soon as the tank became full.

Although this may be rather a long term view, the possibility of these developments occurring should not be overlooked. Obviously vehicle manufacturers would be loath to add to their cars yet another complication, in the form of special vents and seals over the filler apertures, so it is a development that is most likely to be introduced as an anti blow-back device by the accessory manufacturers supplying the retail trade.

The lines of the ADO 15 are highly functional and there is a complete absence of concession to the styling tricks now in favour. Although the car is a genuine four-seater, its overall length is almost exactly 10 ft, and overhang at both ends is small

B.M.C. ADO 15

Part I: The 848 cm³ Four-Cylinder Engine, the Gearbox and the Final Drive to the Front Wheels

SINCE about the middle of the last decade, the European concept of the so-called people's car has, by the familiar processes of evolution, passed from the bubble car to the genuine small car of very compact design. Although they are thoroughly practicable in most respects, these small cars tend to have one or both of two major shortcomings: the first is inadequacy of the accommodation, in terms of the comfortable conveyance of four adults, and the second is a relatively low power : weight ratio, which restricts their acceleration, particularly when a full load is carried. Consequently, hard driving is necessary to keep up with the traffic stream, and this in turn leads to undesirably high rates of wear. Few small engines, too, can be driven at high speeds and loads without being noisy.

When The British Motor Corporation decided some four years ago that a large potential market existed for a smaller car than its Morris Minor 1000 and Austin A.35 models, the design team's terms of reference covered accommodation for four full-size adults and a performance at least equal to that of these existing vehicles. It was also stipulated that there should be adequate stowage space for luggage and other articles; the overall length should not exceed 10 ft, all wheels should be independently sprung, and the fuel consumption under normal driving conditions should be substantially in excess of 40 m.p.g.

Here, clearly, was a formidable list of requirements, which could only be satisfied by major departures from orthodox practice. The challenge was met in the most able fashion, with the result that the ADO 15 car—produced in two almost identical versions, as the Austin Seven and the Morris Mini-Minor—has already become acknowledged as a leader in its class and may well have started a new trend. While the efforts of the design staff deserve high praise, credit must be given also to the B.M.C. management for approving the production of such an unconventional design. Full justification of this initiative is afforded by the fact that, at the time of writing, over 160,000 units had been sold since the cars were introduced during the summer of 1959.

The most outstanding single feature of the vehicle is undoubtedly the placing of the 848 cm³ four-cylinder power unit transversely at the front, driving the front wheels. In this way, the longitudinal space requirement of the engine is no greater than that of a conventionally installed opposed twin-cylinder unit, thereby leaving available for the occupants almost the entire space between the wheel axes. The transverse crankshaft layout is, of course, not new: it is also used in the case of the parallel-twin-cylinder engine of the N.S.U. Prinz, introduced a year earlier than the ADO 15. However, the B.M.C. application appears to be the first, at least in modern times, to feature a frontally mounted, four-cylinder engine. The radiator is mounted at the left-hand end of the unit.

Having decided on a transverse, frontally installed engine, the B.M.C. design team was then faced with the problem of evolving a satisfactory layout for the transmission. The solution to this problem is particularly ingenious. Since the available width was insufficient for the gearbox mainshaft to be in line with the crankshaft, the drive had to be turned through an angle; the simplest and most compact assembly was obtained by making this angle 180 deg. However, as a further means of achieving maximum passenger space, with minimum frontal overhang, the gears and selector mechanism are carried in the engine sump, not in a separate casing. The sump, incidentally, was originally a magnesium alloy casting, but commercial considerations dictated a change to aluminium.

Because of the need to reverse the drive through 180 deg, the clutch—instead of being of the conventional through-drive type—is of the design commonly employed on motorcycles, in which the driven member is attached to a sleeve embracing the driving shaft. In the ADO 15 layout, the clutch centre is mounted on the end of the crankshaft, and the driven member carries a spur gear at its inboard end. This gear meshes with an idler gear, which in turn drives another gear on the input shaft of the gearbox, thereby bringing the mainshaft clear of the crankshaft assembly.

Since the differential output shafts of the final-drive unit

are parallel with the gearbox mainshaft, the customary crown wheel and pinion are replaced by a pair of spur gears: the pinion is mounted on the mainshaft, and the large gear in a casing bolted to the rear face of the sump. The drive is transmitted to the front wheels by two half-shafts, universally jointed at each end; a splined, sliding coupling is embodied in each inboard joint. Because of the inevitable lack of torsional flexibility in a front-wheel-drive system, these inboard joints have rubber elements; the outboard joints are of the constant-velocity type, owing to the relatively large angles of articulation involved.

It is generally thought that resonant vibrations of the body structure are more difficult to avoid with a transverse engine installation than where the crankshaft is longitudinal. According to B.M.C, however, the magnitude of this problem has been over-rated, certainly in the case of small engines, though care is necessary in the selection of the method of engine mounting. In Part II of this article, details will be given of the mounting system employed on the ADO 15 vehicles. The engine and final-drive unit are carried on a sub-frame, which is attached to the front end of the body shell and also provides the anchorage points for the independent suspension.

Geometrically, the front suspension is orthodox, since it is of the familiar double transverse link type, but it is remarkable for the use of variable-rate rubber springs. A rack and pinion steering system is employed; the rack unit is mounted on the body structure instead of on the sub-frame, in order to simplify the removal of the complete frontal assembly for major servicing or repairs.

The design of an independent suspension system for the undriven rear wheels presented no great difficulty. A simple trailing arm layout is employed and, again, rubber is the springing medium. The arms are pivoted on a second sub-frame, which incorporates anchorages for the spring units.

In the body structure, normal B.M.C. practice is followed, with one noteworthy exception. This is the use of externally flanged joints between the main sub-assemblies. The principal object of using such joints is to facilitate production, but a virtue has been made of necessity, in giving them a minor styling function. Fortunately, though, the stylists have not been permitted to make the lines of the body conform to the current vogue, which would not suit so small a vehicle. In spite of its very functional shape—or perhaps because of it—the car has a neat and purposeful attraction of its own.

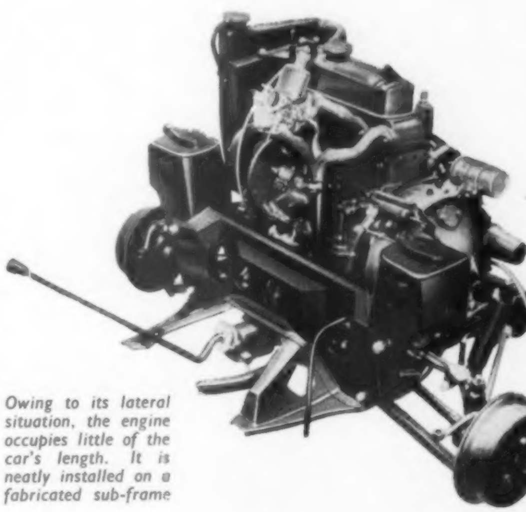
Mention was made earlier that the ADO 15 body is a full four-seater. This is no mere sales propaganda, for the passenger space is as great as that of the Morris Minor 1000; in fact, certain internal dimensions are greater than their equivalents in the larger car, although the overall lengths differ by as much as 28 in. No less remarkable is the amount of stowage space for luggage, parcels and other impedimenta. The boot is of only moderate capacity but its lid is hinged at the bottom and adequately supported to serve as a platform for the carriage of additional baggage. There are also large pockets in the doors, others alongside the rear seat and beneath its leading edge, and the usual shelf between the squab and the rear light. Finally, the effective width of the parcels shelf on the dash is greater than average, since it is interrupted only by the speedometer, which is situated medially to suit both right-hand and left-hand drive.

It might be thought surprising, in view of the ultra-modernity of so much of the car, that quite extensive routine lubrication of the chassis is necessary. The reason for this is that the Corporation, in spite of its extensive investigation of bearings not requiring regular lubrication, is still not satisfied with the reliability of the available sealing arrangements. Some simplification of servicing results, however, from the fact that the gearbox and final-drive unit share a common oil supply with the engine.

Because of the transverse, frontal disposition of the engine, accessibility is excellent for such work as decarbonizing, or checking the valve clearances or contact breaker gap. The rearward side of the unit is no less accessible than is either side of most conventionally mounted engines. As has already been mentioned, the complete sub-frame, engine, transmission and suspension assembly can readily be detached from the vehicle for any work of a major nature.

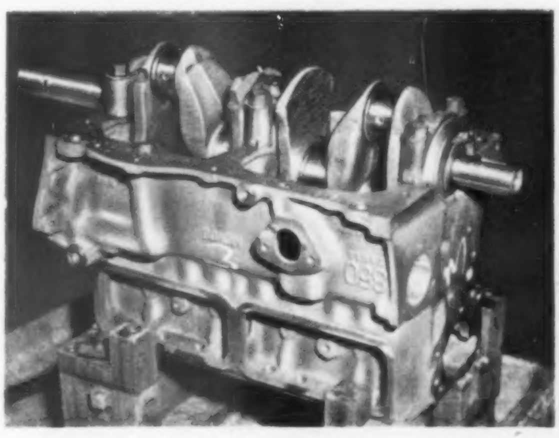
Although the power unit is very near the front of the vehicle, the installation is considered to be less vulnerable to frontal impact than is one with a conventional longitudinal disposition of the engine. It is almost unknown, with any car, for the main structure of the engine to be broken in a collision, and a smashed distributor is cheaper to replace than is a damaged radiator. There has been some criticism of the susceptibility of the ignition system to water thrown up by preceding vehicles, but this matter is receiving attention.

The first extension of the Austin and Morris ranges of the ADO 15—which originally comprised only standard and de luxe saloons—came in August 1960, with the introduction



Owing to its lateral situation, the engine occupies little of the car's length. It is neatly installed on a fabricated sub-frame

The cylinder block casting of the Series A engine is employed also for the ADO 15 power unit, but the crankshaft has a shorter throw and its rear end is longer, to suit the different clutch layout



of a 5 cwt van of both makes. Apart from the obviously necessary differences in the body, the van version has a wheelbase 4 $\frac{5}{16}$ in longer than that of the car, in order to provide the desired floor area for the carriage of goods. Because of the low, flat floor permitted by the front wheel drive, the overall height is virtually the same as that of the car; the capacity of the body is normally 46 ft^3, but it can be increased to 58 ft^3 if the passenger's seat is removed.

During September 1960, an estate car version was added to each range. The Austin Seven model is called the Countryman and the Morris variant the Traveller; both terms have already been used by B.M.C. for this type of body on larger vehicles. As would be expected, these estate cars have the longer wheelbase. Consequently, the passenger accommodation is almost identical with that of the normal saloons, but the capacity for carrying luggage or goods is considerably increased. In the manner customary for such vehicles, the squab of the rear seat folds forward to permit the carriage of really bulky or awkwardly shaped items.

Engine

The engine of the ADO 15 has much in common with the B.M.C. Series A 948 cm^3 unit, which has given such good service in the Austin A.35 and A.40 and Morris Minor 1000 vehicles. Since this larger engine, in the form employed in the Austin-Healey Sprite, was described in the July 1960 issue of *Automobile Engineer*, a detailed description of its derivative is hardly necessary. Instead, its basic layout will be reviewed and reference will be made to certain noteworthy features and to the differences from the Series A power unit.

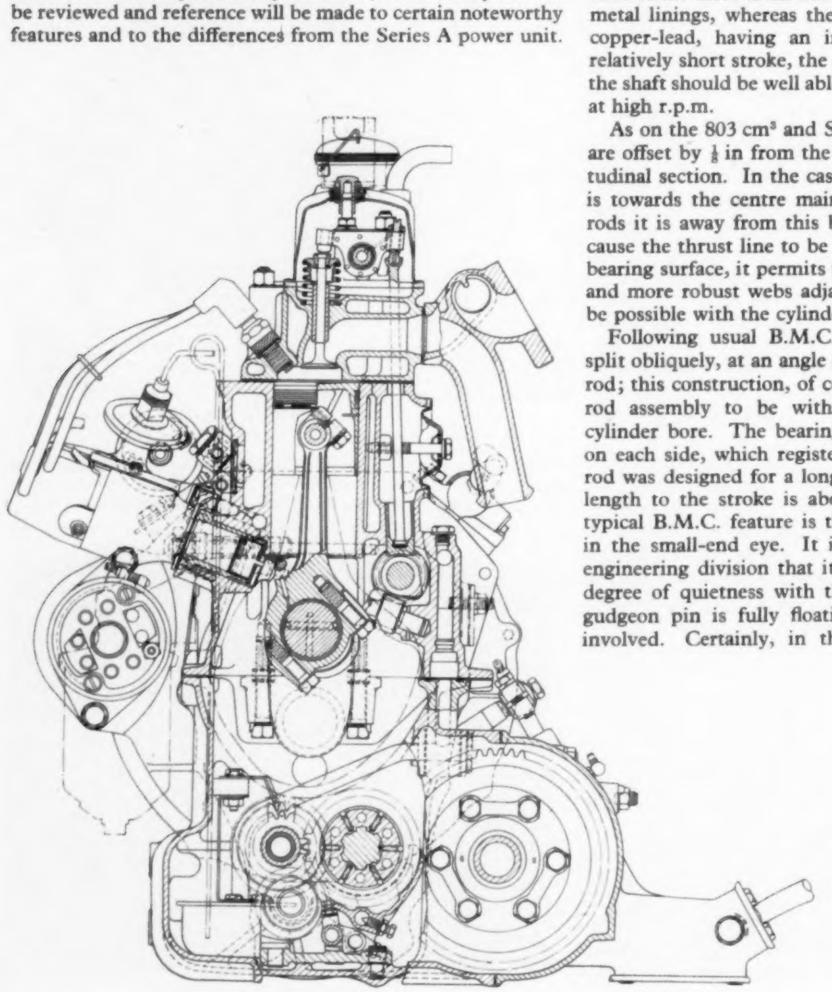
First, however, it should be mentioned that the Series A engine was evolved from the 803 cm^3 unit fitted to the original post-war Austin A.30 car. This smaller engine had a stroke of 76·2 mm, which was retained for the Series A, but its bore was only 58 mm, giving the rather high stroke: bore ratio of 1·31:1. The increase of bore to the 62·9 mm of the Series A unit was made without increasing the external length of the block, by siamesing the front and rear pairs of cylinders; the water space between cylinders 2 and 3 was retained. To withstand the higher power output and torque that resulted from the greater swept volume, the crankshaft and connecting rods were strengthened.

For the ADO 15 engine, the Series A block and head are employed without alteration. However, the swept volume has been reduced to 848 cm^3 by fitting a new crankshaft having a stroke of 68·26 mm. The resultant stroke : bore ratio of 1·08:1 is more in line with current thought on this aspect of design than was the Series A figure of 1·21:1. As a further means of rationalizing production, the Series A crankshaft bearings and connecting rods have been adopted; the appropriate clearance volume is obtained by the use of pistons that are higher above the gudgeon pin axis than are those of the 948 cm^3 power unit.

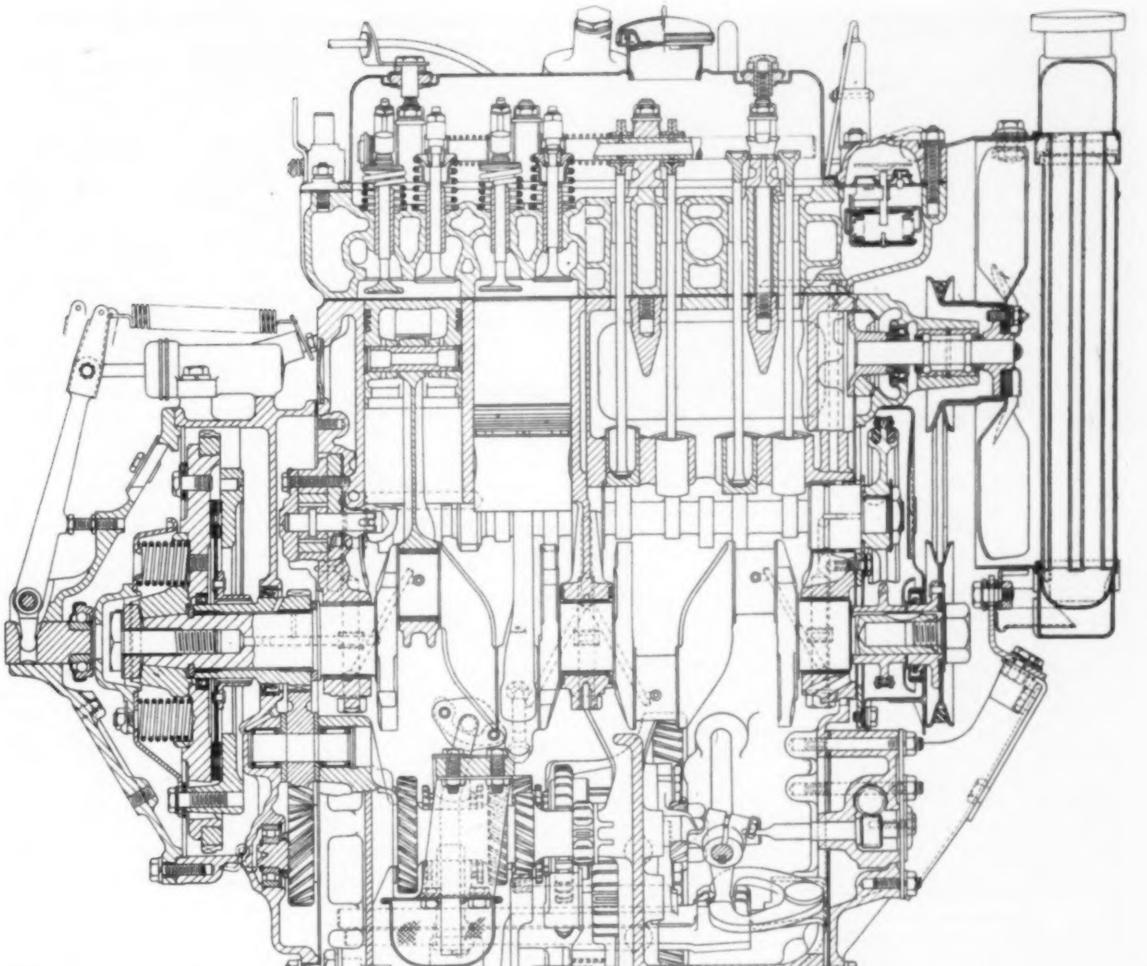
Regarding the basic design, the block is an orthodox iron casting, and the joint face with the sump is on the horizontal plane through the crankshaft axis. The forged crankshaft embodies integral counterweights on the webs adjacent to each of the three main bearings. These bearings have white-metal linings, whereas the big-end bearings are lined with copper-lead, having an indium overlay. Because of the relatively short stroke, the journal overlap is considerable, so the shaft should be well able to withstand sustained operation at high r.p.m.

As on the 803 cm^3 and Series A engines, the big-end eyes are offset by $\frac{1}{8}$ in from the cylinder axes, as viewed in longitudinal section. In the case of cylinders 1 and 4, the offset is towards the centre main bearing, and on the other two rods it is away from this bearing. Though this offset does cause the thrust line to be displaced from the middle of the bearing surface, it permits the use of a wider centre bearing, and more robust webs adjacent to it, than would otherwise be possible with the cylinder spacing chosen.

Following usual B.M.C. practice, the big-end eyes are split obliquely, at an angle of 45 deg to the major axis of the rod; this construction, of course, is to enable the piston and rod assembly to be withdrawn or inserted through the cylinder bore. The bearing cap is located by tongues, one on each side, which register in slots on the rod. Since the rod was designed for a longer-stroke engine, the ratio of its length to the stroke is above average, at 2·14:1. Another typical B.M.C. feature is the clamping of the gudgeon pin in the small-end eye. It is the opinion of the company's engineering division that it is easier to ensure a reasonable degree of quietness with this arrangement than where the gudgeon pin is fully floating, since only one clearance is involved. Certainly, in the case of the various B.M.C.



This transverse sectional view of the ADO 15 engine shows how the gearbox shafts are situated below the crankshaft. The final-drive reduction is effected by means of a pair of helical spur gears. Among the other noteworthy features of the engine are the use of combined inlet and exhaust manifolds and of fabricated type valve rockers



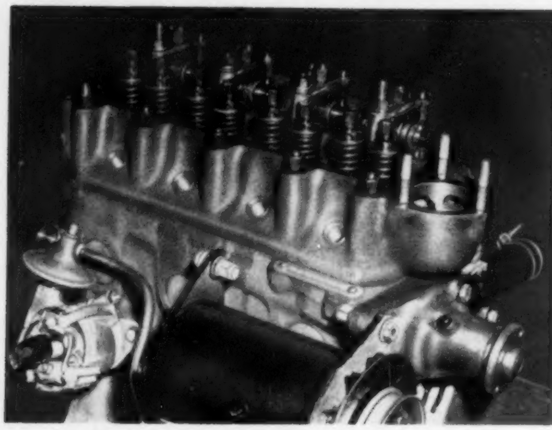
Above the crankshaft, the engine differs from the Series A unit only in respect of the pistons, which are taller above the gudgeon pin, to keep the compression ratio at 8·3 : 1. The clutch is mounted on an extension of the crankshaft; its output member is splined to a sleeve that floats on the shaft and embodies a helical gear. An idler gear is interposed between this gear and that mounted on the first-motion shaft of the gearbox assembly

engines, wear between the pin and the piston is no more pronounced than in fully floating layouts.

The side mounted camshaft is driven from the front end of the crankshaft by a single-row roller chain of $\frac{3}{8}$ in pitch. At the front, the camshaft is carried in a steel backed white-metal bearing, but the other two bearings are machined directly in the block. The rear end of the shaft is counterbored to receive the slotted spindle of the Hobourn-Eaton lobe type oil pump, which is driven by means of a peg. This pump is primed initially, through a hole in the side of the block casting, and thereafter retains sufficient oil when at rest to be self-priming. A nominal capacity of 8 pints is quoted for the sump which, as already stated, also serves the gearbox and final-drive unit.

The camshaft actuates the solid push rods through the medium of bucket type tappets, which run directly in the block. Oil is fed to the camshaft front bearing, and thence, through internal passages in the block and head, to the hollow rocker shaft, which is mounted in four horizontally split pedestals, and has radial drillings for the lubrication of the rocker bearings. Unusally, the rockers are not stampings, but each consists of two 11 s.w.g. steel pressings. One pressing is the mirror image of the other and the two are joined by projection and butt welding. Each incorporates a flanged hole that forms half of the bearing housing. The rocker bearings are Clevite 8 wrapped copper-lead bushes. Internal passages take the oil from the bearing surface of each rocker to the ball face that seats in the push rod cup end. These rockers, which are made by Vandervell Products Ltd, are both lighter and cheaper than were the stampings used on the A.30 engines and on the earlier Series A units, which are now fitted with the pressed components.

Dimensions and other particulars of the valves and springs are quoted in the accompanying table. The valves are installed vertically and in line in the cast iron cylinder head. Oil loss down the valve guides is minimized by means of chamfers on the upper ends of the guides, shrouds round the upper portions of the valve stems, within the springs, and rubber O-rings below the collets. All the ports are on the rearward side of the engine, as installed, and the inlet ports of cylinders 1 and 2, and those of cylinders 3 and 4, are siamesed. This commonly employed arrangement does



The thermostat is housed conventionally, at the front of the cylinder head. Each overhead rocker comprises two steel pressings, which are welded together and house a Clevite 8 wrapped type copper-lead bush

simplify the manifolding and, although in theory it tends to give a lower volumetric efficiency than does the use of separate ports, it is the opinion of B.M.C. that, in the case of an engine with one carburettor, this disadvantage is offset by the improved distribution. Siamesing is also employed for the exhaust ports of cylinders 2 and 3.

The combustion chamber is, of course, of the heart shape developed by Harry Weslake from the orthodox bath tub design. A compression ratio of 8·3:1 has been adopted, which is the same as the higher of the two ratios available in the Series A engine. Because of the usual scale effect, the ADO 15 unit has, in fact, a lower octane requirement than has the 948 cm^3 model on that ratio. The valves are common to both engines: since the valve lift is also the same, the ratio of opening area to swept volume is about 12 per cent greater in the case of the smaller unit.

A combined inlet and exhaust manifold casting was developed for the ADO 15 and has subsequently been adopted also for the Morris Minor 1000, which likewise has an S.U. carburettor. The casting is of iron and embodies an effective hot spot, consisting of a common wall at the middle of the gallery of the inlet portion. A downdraught angle of 30 deg is given to the S.U. HS2 instrument. To the intake of the carburettor is bolted an aluminium elbow that connects it to the paper element air filter. The carburettor is fed by an S.U. PD electric pump mounted on the rear sub-frame.

Bolted to the left-hand end of the sump is a bracket fabricated from 10 s.w.g. steel. This bracket carries one of the rubber block mountings of the power unit—to be described in Part II of this article—and it also forms the lower support of the radiator, the top of which is attached to the cylinder head by a second bracket. To minimize the effects of high-frequency vibrations, rubber washers are embodied in the radiator mounting. The capacity of the engine cooling system is 5¼ pints, which is increased to 6¼ pints when the heater is fitted.

Encircling the four-blade fan, to increase its efficiency, is a shallow duct of sheet steel, which is integrated with the radiator shell. Air enters the engine compartment through the normal frontal opening in the bonnet, is blown through the radiator by the fan and is exhausted through a grille in the inboard panel of the left-hand wheel arch. The flow of air is assisted by the fact that an appreciable depression exists under the front wings when the vehicle is in motion. To overcome criticism of noise at high engine speeds, a new design of fan with a larger number of blades has been evolved and is going into production.

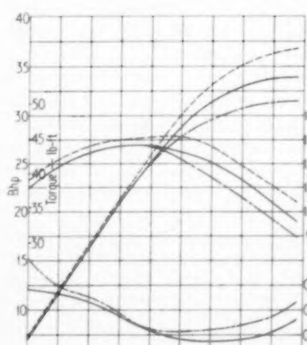
Curves for the engine performance, measured at the flywheel, are reproduced in an accompanying illustration. Those shown by full lines are for the engine to B.M.Cs standard test specification. The gross performance, shown by dotted lines, is measured under conditions similar to standard except that the air filter is removed and manual controls are employed for the carburettor and ignition. Chain dotted lines are used to indicate the performance as installed; the engine specification then differs from that for the standard curves in that the cooling fan is fitted, and the car's exhaust system replaces that of the test bed installation.

It is of interest that the gross power output of 37 b.h.p. at 5,500 r.p.m. is slightly greater than that of the Austin A.40—which has a Zenith carburettor and produces 36 b.h.p. at 4,800 r.p.m.—and less than that of the Morris Minor 1000, which is 40 b.h.p. at 5,000 r.p.m. The higher specific output of the ADO 15 unit, 43·7 b.h.p/litre as against 38·0 for the A.40 and 42.2 for the Minor, is mainly attributable to the proportionately larger valves, which have made possible an upward extension of the speed range. Whereas the respective maximum piston speeds of the two 948 cm^3 engines are 2,400 and 2,500 ft/min, that of the ADO 15 unit is 2,460 ft/min.

As would be expected, however, this virtual parity of top end performance is not maintained in terms of maximum torque, the figures for which—in the same order as before—are 50 lb-ft at 2,200 r.p.m, 50 lb-ft at 2,500 r.p.m, and 44 lb-ft at 2,900 r.p.m; the effect of the larger ratio of valve area: swept volume on the speed is considerable. The ADO 15 has a maximum gross b.m.e.p. of 132 lb/in², about average for an engine of this type, and its b.h.p/in² of piston area is 1·92. Because of the incorporation of the gearbox and final-drive unit, the power: weight ratio of 0·12 b.h.p/lb is clearly not directly comparable with that of most other engines. The ratios of maximum torque: torque at maximum

A single iron casting forms both manifolds; to provide a hot spot, there is a common wall between the portions. It is now used on the Morris Minor 1000 car as well as the ADO 15

Performance curves for the ADO 15 engine. An explanation is given in the text regarding the conditions of test applying in each of the cases



Valve data—PRINCIPAL DIMENSIONS AND SPECIFICATIONS

	Inlet	Exhaust
Material	En.52	En.59
Head diameter	$1\frac{3}{32}$ in	1 in
Throat diameter	$\frac{15}{16}$ in	$\frac{29}{32}$ in
Stem diameter	0·2793 to 0·2798 in	0·2788 to 0·2793 in
Diametral clearance in guide	0·0015 to 0·0025 in	0·002 to 0·003 in
Seat angle	45 deg	45 deg
Face width—on valve	0·070 in	0·070 in
on seat	0·066 in	0·088 in
Spring material	En.49D	
Wire diameter	0·136 in	
Coil outside diameter	$1\frac{1}{8} \pm 0·007$ in	
Number of effective coils	$4\frac{1}{2}$	
Free length	$1\frac{5}{8}$ in	
Installed length	$1\frac{11}{32}$ in	
Spring rate	104 lb/in	
Valve lift	0·281 in	
Operating clearance, at valve	0·012 in (cold)	

power and of speed at maximum torque: speed at maximum power are respectively 1·24:1 and 0·527:1.

Although, because of their low weight and small frontal area, these vehicles give an excellent fuel consumption on the road, there is nothing remarkable about the specific consumption as shown by the curves. The standard curve has a minimum of 0·54 pt/b.h.p-hr, at about 4,000 r.p.m, and the consumption in the installed condition has its lowest point at just over 0·55 pt/b.h.p-hr, at about 3,500 r.p.m. However, the curves are satisfactorily flat, as is demonstrated by the fact that the last mentioned one lies below 0·6 pt/b.h.p-hr between 2,400 and 5,200 r.p.m.

Clutch

Because of the unusual design of the clutch, it is largely manufactured by B.M.C, though a Lockheed hydraulic actuating cylinder and a Newton and Bennett $7\frac{1}{4}$ in diameter driven plate are employed. The flywheel centre, a forging of En.3 steel, is mounted on a taper on the end of the crankshaft, on to which it is pulled by a $\frac{5}{8}$ in diameter bolt screwed into the shaft. To locate the flywheel so that the timing marks on it are positioned appropriately relative to the crankshaft, both the centre and the shaft have a diametral slot machined in their end faces; beneath the head of the securing bolt is fitted a special washer having a corresponding diametral tongue, or key, which registers in both slots. The bolt is locked by means of a tab washer.

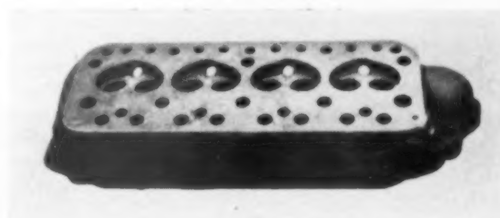
Four $\frac{3}{8}$ in bolts attach the flywheel—which serves also as the clutch reaction plate—to the centre; location is effected by means of a spigot. The flywheel is of cast iron, is 10·2 in diameter and has the usual Bendix toothed ring shrunk on to its periphery. It is of interest that, originally, the flywheel and its centre comprised a single iron casting, but it was felt that a composite construction would better withstand the bursting forces involved. Immediately inboard of the flywheel is the driven plate, which is of the usual laminated steel construction and carries a riveted-on friction ring, nominally 0·1 in thick, on each side. This plate is riveted to its hub, which is internally splined; these splines are a sliding fit on the external splines of a sleeve gear floating on the crankshaft.

The cast iron pressure plate is, of course, situated inboard of the driven plate; it has three bosses that project axially from its periphery and pass through slotted holes in the flywheel. To the bosses is bolted the cover, a 12 s.w.g. domed, steel pressing of approximately triangular shape, with a peripheral lip. The securing bolts also anchor the three spring steel links whereby the reaction plate is connected to the pressure plate. This arrangement is similar to that employed on the Borg and Beck strap drive clutches: since it permits the necessary relative axial movement without the use of parts that can wear, it ensures quiet operation of the clutch over long periods.

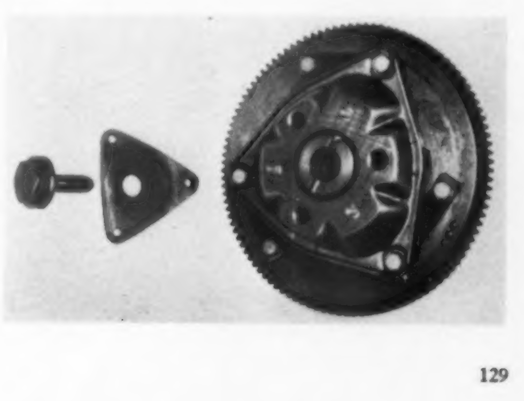
Six helical springs apply the clutch clamping pressure. They have the following particulars: diameter of wire, 0·128 in; coil outside diameter, 1·128 in; number of effective coils, 7 to $7\frac{1}{2}$; free length, 2·682 in; installed length, clutch engaged, 1·402 in; total rate, 309·6 lb/in; total thrust, clutch engaged, 420 lb; total thrust, clutch released, 445 lb. At their inboard ends the springs seat in blind holes in the clutch centre, the holes alternating with the attachment bolts. The other ends of the springs are located by internal bosses in the cover.

Completing the clutch assembly is the forged steel dome to which the release thrust is applied. It is secured to the cover by three studs passing through its flange, and is located by means of a spigot that enters the hole in the middle of the cover. A simple actuating mechanism has been adopted, comprising a rocking lever, with a short and a long arm, and a plunger carrying a special Ransome and Marles ball thrust bearing. The lever is pivoted on the clutch housing, adjacent to the central boss in which the plunger is a sliding fit. Whereas the short arm of the lever registers in the plunger, the long one is connected by a clevis to the piston rod of the hydraulic actuating cylinder. Movement of the piston therefore produces an axial displacement of the plunger, bringing the bearing into contact with the thrust dome.

The plunger is of case-hardened En.32 steel, and the register for the arm—which has a hardened ball end—is in the form of a diametral hole, chamfered to provide clearance for the angular movement. At its inboard end, the plunger is shouldered to take the inner race of the ball bearing, which contains eight $\frac{9}{32}$ in diameter balls. On the outer race of the bearing is an inward-projecting flange that forms the thrust face. A light-gauge steel pressing forms a dust shield over



Above: The combustion chambers are of the familiar Weslake heart shape. Both pairs of inlet ports are siamesed. Below: Six springs are incorporated in the clutch; the reaction and pressure plates are connected by the three straps seen here



the outboard end of the bearing; it is peened over the outer race and there is a running clearance between it and the inner race.

Two aluminium alloy castings comprise the bell housing and the case for the gear drive to the primary shaft of the gearbox. The first casting has the approximate shape of a short cylinder and is bolted to the rear of the block and the sump, in the position occupied by the normal bell housing of the Series A engine. The casting is divided into two compartments by a diaphragm type wall, on one side of which is the gear train and on the other the clutch. A double lip oil seal, carried in the wall and bearing on the crankshaft sleeve gear, prevents any transfer of oil or friction material dust from one side to the other.

The second casting, of conical shape, forms the clutch housing and is located on the first one by a circumferential spigot; it is secured by a ring of ¼ in bolts. In it is an inspection hole, for checking the ignition timing, and it embodies a boss into which is screwed the stop bolt for the actuating lever. There are also two tapped bosses, below the clutch axis, for the attachment of the right-hand rubber mounting.

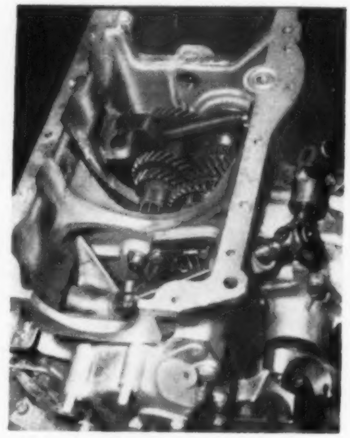
Primary gear train and gearbox

Details of the gears used in the primary train, and of those of the mainshaft and layshaft clusters, are given in the accompanying table. Reference has already been made to the fact that the sleeve gear on the crankshaft is externally splined to take the drive from the clutch driven plate carried on it. The sleeve is located axially on the shaft between a thrust washer and a clip of horseshoe shape: the thrust washer seats against a shoulder adjacent to the rear main bearing, and the clip fits into a groove in the shaft immediately ahead of the flywheel centre. A shallow counterbore in the centre overlaps the clip and so retains it in the groove.

When the vehicle is in gear and the clutch is disengaged, or partially engaged, there is, of course, a speed differential between the crankshaft and the sleeve gear. Because of this relative rotation, two plain bearings are interposed between the two components. These bearings, one of which is pressed into the bore of the sleeve gear at each end, are of the standard steel backed, wrapped, white-metal type and have a nominal bore of 1⅜ in. The one nearest the rear main bearing is ¾ in long and the other has a length of ⅝ in.

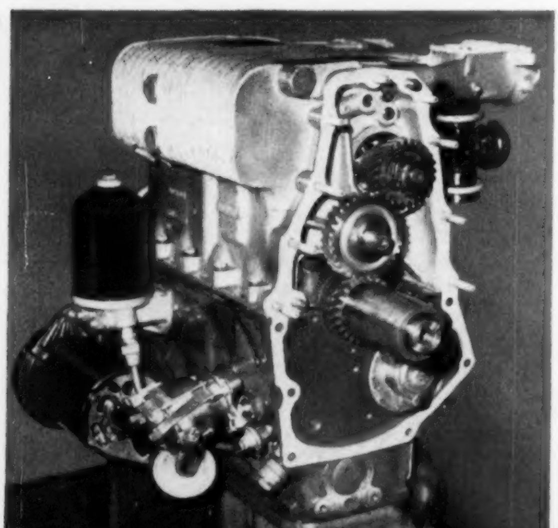
From a drilling in the rear journal of the crankshaft, an oil feed is taken to the nearer of the two bearings. The other bush is lubricated by seepage along the bore of the sleeve, not by pressure, because it is the more lightly loaded of the two and a pressure feed would introduce sealing complications in respect of the clutch. To relieve the pressure, the oil emerging from the first bearing is exhausted through relief holes in the sleeve, between the two bearings, and drains back to the sump. Whatever oil may exude from the outboard end of the second bearing is prevented from reaching the clutch by a lip type seal that is housed in the bore of the flywheel and bears on the exterior of the sleeve.

The idler gear of the primary train is pressed on to a ¾ in diameter live spindle that runs in two Torrington needle roller bearings, to avoid the need of pressure lubrication. One of these bearings is housed in a boss in the partition wall of the bell housing front casting, and the other in a boss in the sump. On each side of the gear is a heat treated En.18C thrust washer. The bearing in the sump is lubricated by splash from the gearbox, and the other by oil thrown from the primary train, by way of an inclined passage drilled in the bearing boss.



The layshaft is sited ahead of the mainshaft, and the bearings for both are installed in webs across the sump. An oblique disposition of the striker shaft is necessary to bring the linkage above the final-drive assembly

The idler gear of the primary train is larger than the other two gears; it has a live spindle, which is carried in two needle roller bearings



A roller bearing of 1$\frac{9}{16}$ in overall diameter and ⅝ in bore, housed in the partition wall, carries the right-hand end of the first-motion shaft of the gearbox, which is splined to accept the driven gear of the primary train. The outer race of this bearing is located against a shoulder in the housing boss, and the inner race against a shoulder on the shaft; both are retained by circlips. Supporting the other end of the shaft, immediately adjacent to the constant-mesh pinion, is a ball bearing, which has an outside diameter of 2½ in, a bore of 1 in and an axial length of ¾ in. To provide axial location of this second bearing in its housing in the sump wall, its outer race has a flange that butts against a shoulder in the housing, in which the race is secured by a circlip. The inner race is trapped against a shoulder on the shaft by an extension of the boss of the driven gear of the primary train.

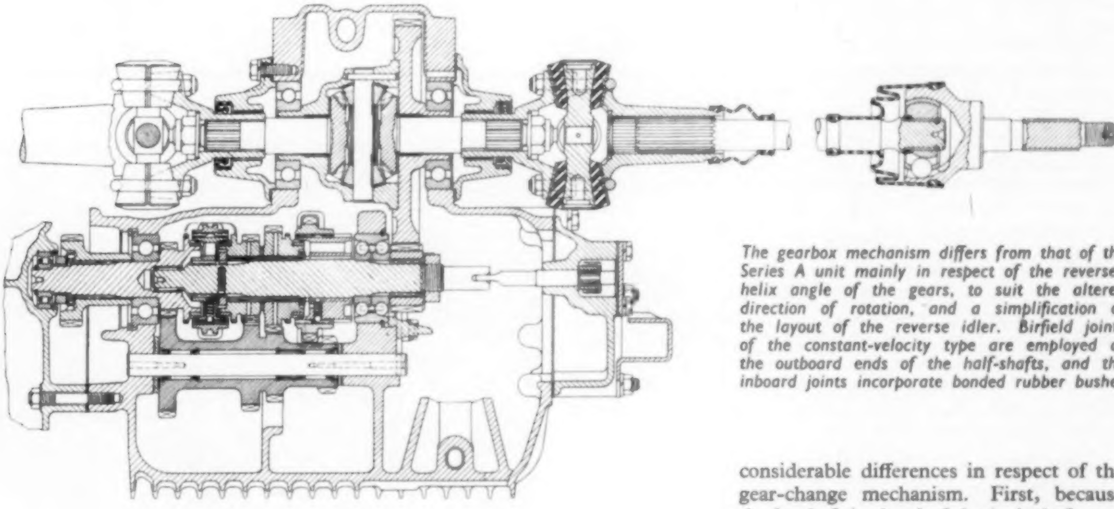
The gearbox has synchromesh on the upper three forward ratios, and sliding pinion engagement of first and reverse gears. Its mechanism is based on that of the current Austin A.40 and Morris Minor 1000 vehicles. However, because the final-drive pinion is mounted directly on the mainshaft, this shaft and its bearings are of different design. By com-

parison with the synchromesh of the Series A gearbox, that of the ADO 15 has to cope with a system of higher inertia, resulting from the larger clutch and the primary gear train.

Owing to the reversed direction of rotation of the gears, compared with that of the Series A assembly, the helix angle of the gears is of the other hand. In addition, the spacing of the ratios has been improved by raising that of second gear by about 9 per cent, from 2·375:1 to 2·172:1. The other ratios are as follows: top, 1:1; third, 1·412:1; first and reverse, 3·628:1. To reduce gear selection difficulty, the reverse idler layout has been altered so that only one pair of gears is brought into mesh when the first ratio is engaged.

purpose for the mainshaft bearing. Since the needle rollers are uncaged, each set is located by a bronze and steel thrust ring at each end. The rings are a sliding fit in the bore of the cluster, in which they are retained by circlips. There is another thrust ring between each end of the cluster and the face of the adjacent supporting web. Lubrication of the needle roller bearings is by splash: an axial hole is drilled inward from each end of the spindle, and at its inboard end is a downward facing radial hole positioned mid-way between the ends of the rollers.

In most other respects, the layout of the actual gearbox is similar to that of the Series A unit. However, there are considerable differences in respect of the gear-change mechanism. First, because the layshaft is ahead of the mainshaft, and not below and to one side of it, the selector rods lie directly beneath the gear assembly, instead of being offset to the other side. Moreover, the connection between the gear lever and the striker is necessarily indirect, since the final-drive unit is situated between the gear assembly and the driver of the vehicle. The striker shaft is situated above the mainshaft; it is disposed longitudinally as viewed in plan but is inclined appreciably upward towards the rear. Its trailing end projects through the rear wall of the sump, and on it is mounted a transverse lever with a ball end; this lever has a split boss and is clamped in position by a bolt that is also a cotter.



The gearbox mechanism differs from that of the Series A unit mainly in respect of the reversed helix angle of the gears, to suit the altered direction of rotation, and a simplification of the layout of the reverse idler. Birfield joints of the constant-velocity type are employed at the outboard ends of the half-shafts, and the inboard joints incorporate bonded rubber bushes

SAE8615 case-hardening steel is employed for the mainshaft, the maximum diameter of which—over the splines for the first and second gear dog member—is 1¼ in. It is supported at its left-hand end in a special double-row ball bearing, which has separate inner races but a single outer race. The overall diameter of this bearing, and its bore and axial length—over the inner races—are respectively 2½ in, 1 in and 1 in. The inner races are held between a shoulder on the mainshaft and the final-drive pinion, which is secured on its splines by a nut and tab washer. Location of the outer race in its housing, which is in a transverse web within the sump, is effected by means of a circlip in a groove around its periphery; this circlip registers in a counterbore. A small retainer bracket, bolted to the web, bears on the end of the outer race to clamp it in position.

Carrying the other end of the mainshaft is a Torrington needle roller bearing, the outer race of which is pressed into a counterbore in the primary shaft. The advantages of the Torrington bearing over a plain bush in this application are, of course, that it has a high load carrying capacity in relation to its overall dimensions, and that no special provision has to be made for lubrication.

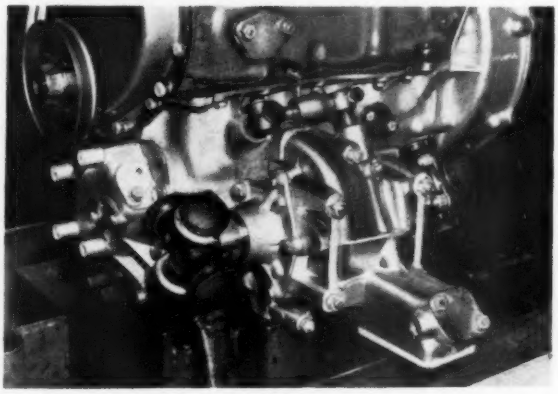
As on most other gearboxes of this type, all the gears on the layshaft are integral, and the unit revolves on a fixed spindle. Near each end of the bore of the layshaft cluster is a needle roller bearing, comprising 23 rollers 2½ mm diameter × ⅝ in long. These rollers run directly in the bore and on a ⅝ in diameter hardened spindle. The spindle is a press fit in two webs in the sump; one of these webs is that which also houses the bearing of the first-motion shaft, and the other has already been mentioned as serving the same

The ball end registers in a socket machined in a second lever, which is clamped on to the upper extremity of an intermediate shaft, inclined downward and rearward at right angles to the striker shaft. Only the upper end of the intermediate shaft protrudes from the final-drive casing, in which the shaft is carried in two plain, porous bronze bearings.

At the bottom end of the shaft, immediately below the lower bearing, is a short integral arm having a ball socket at its extremity. This arm operates in an extension of the final-drive housing. In the socket fits the ball end of the gear lever, which has a conventional spherical mounting in a steel cover plate bolted to the open rear end of the extension. Since the gear lever is inclined at about 45 deg, its use necessitates unusually oblique movements of the hand and arm, but a driver quickly becomes accustomed to this. Though the lever is relatively long, the leverages have been arranged so that the travel between the various gear positions is comfortably short. Earlier criticisms of the gear-change have been overcome by the fitting of a stronger reverse baulk spring and by increasing the clearances in the linkage.

Gear data—THE PRINCIPAL DIMENSIONS AND SPECIFICATIONS OF MATERIALS

	Number of teeth	Diametral pitch	Helix angle	Material	Gear blank thickness at p.c. diameter
Primary train					
sleeve gear on crankshaft	24	10 T.D.P.	30 deg left-hand	SAE8615	$\frac{5}{8}$ in
idler	31	10 T.D.P.	30 deg right-hand	SAE8615	0·695 in
gear on first-motion shaft	24	10 T.D.P.	30 deg left-hand	SAE8615	$\frac{5}{8}$ in
Mainshaft gears					
first-motion	19	10 T.D.P.	40 deg left-hand	En.361	$\frac{11}{16}$ in
third	23	10 T.D.P.	40 deg left-hand	En.361	$\frac{1}{2}$ in
second	28	10 T.D.P.	40 deg left-hand	En.361	$\frac{1}{2}$ in
first	32	10 N.D.P.	—	En.361	$\frac{7}{16}$ in
Layshaft gears					
constant-mesh	28	10 T.D.P.	40 deg right-hand	En.361	$\frac{11}{16}$ in
third	24	10 T.D.P.	40 deg right-hand	En.361	$\frac{15}{32}$ in
second	19	10 T.D.P.	40 deg right-hand	En.361	$\frac{11}{16}$ in
first	13	10 N.D.P.	—	En.361	$\frac{1}{2}$ in
Reverse idler	18	10 N.D.P.	—	En.18C	$\frac{11}{16}$ in



The casing of the final-drive unit is bolted to the rear face of the sump. When the engine is installed, the blanking plate at the rear of the casing is removed and the gear lever and its mounting are fitted instead

For the speedometer drive of the ADO 15, the left-hand end of the mainshaft has an extension that is diametrically slotted to accept the tongued end of a short spindle. This spindle runs directly in an aluminium casting bolted to the end of the sump, and it carries the drive gear at the end remote from the tongue. The meshing gear is mounted on a spindle running in a plain bearing in a cover plate that is bolted to the aluminium casting.

Final-drive unit

The final-drive pinion has an overhung mounting on the left-hand end of the mainshaft. The pinion has 17 teeth and it meshes with a 64-tooth wheel which, together with the differential unit, is housed partly in the pump and partly in the cast aluminium final-drive casing. This casing is secured by studs and nuts to the vertical rear face of the sump; it is located by two dowels, and the meshing of the gears is controlled by the thickness of the shim washers inserted between the sump and the casing. To give the correct geometry of the half-shafts, the axis of the wheel is $\frac{1}{4}$ in below that of the pinion.

En.361 steel is employed for both the pinion and the wheel, which have a transverse diametral pitch of 10 and a helix angle of 30 deg. The nominal face width of the pinion is $\frac{7}{8}$ in, and that of the wheel is $\frac{3}{4}$ in. Spigoted into the wheel, and secured to it by four $\frac{5}{16}$ in bolts, is the cage of the two-pinion differential, which is the standard B.M.C. Series A assembly described in connection with the Austin-Healey Sprite.

The final-drive unit is carried in the casing in two heavy-duty ball bearings. These have an outside diameter of 3 in, a bore of 1$\frac{3}{8}$ in and an axial length of $\frac{11}{16}$ in. The inner race of the left-hand bearing is pressed against a shoulder on the boss of the large helical gear, and that of the other is similarly mounted on the differential cage. Both outer races fit in half-housings in the sump and the final-drive casing, and are clamped in position when these are bolted together; the races are retained by spigots on the separate housings for the outboard bearings of the differential output shafts. Shims are fitted between the casing and these housings, which are light alloy casings of flanged part-conical shape, to avoid axial pre-loading of the bearings. Each housing is secured to the side of the final-drive casing by five $\frac{5}{16}$ in set-screws.

En.352 steel is used for the differential output shafts, which are integral with their gears. The inboard ends of the shafts are carried in steel-backed white-metal bushes of the wrapped type. One bearing is pressed into the large gear of the final-drive reduction pair and the other into the differential cage; the bearings have a nominal bore of 1 in and are $\frac{5}{16}$ in long.

The outboard portion of each output shaft is splined to accept the yoke of the universal joint, which is clamped against a shoulder on the shaft by a nut and plain washer; the nut is of the self-locking type. Into each of the housings referred to in the penultimate paragraph is inserted a steel-backed white-metal bearing, in which runs the hub of the yoke. These outboard bearings have a nominal bore of 1$\frac{1}{4}$ in and are $\frac{3}{4}$ in long. Outboard of each is a double-lip oil seal that bears on the hub and serves to retain oil and exclude dirt and moisture.

One of the difficulties confronting the designer of a front-wheel-drive car is, of course, that of avoiding transmission harshness. Because of the absence of a long propeller shaft and flexible leaf springs, there is very little inherent torsional flexibility in the system between the engine and the wheels. On the ADO 15, the torsional stiffness of the transmission is reduced by the use of a novel design of inboard universal joint. These joints, which are of the Hooke type, are manufactured by the Dunlop Rubber Co. Ltd, though they were patented and developed by Moulton Developments Ltd, a subsidiary company of B.M.C.

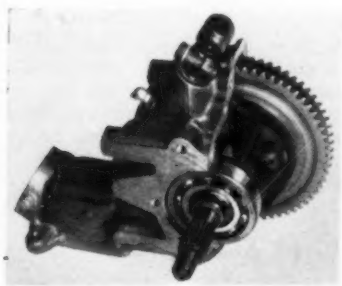
In this joint, the driving and driven yokes, both steel forgings, are connected by a four-arm steel spider, to each arm of which is bonded a rubber bush. The bushes are of part-conical form, with the larger diameter at the greater

radius, and each embodies a circumferential groove. Embracing each bush are two half-shells: these are 14 s.w.g. steel pressings with flanged ends, and each incorporates a semi-circumferential rib or channel that registers in the groove in the bush.

One of the two half-shells for each bush fits into a part-conical recess in the arm of the yoke and the other is encircled by a $\frac{5}{16}$ in diameter U-bolt. The shank of the U-bolt seats in the channel of the half-shell and the ends pass through holes in the yoke. When the stiff-nuts on the U-bolt are tightened, the rubber is compressed until the flanges of the half-shells butt together. Since this precompression increases the load carrying capacity of the joint, the assembly is exceptionally compact for its duty. About 12 deg of relative torsional movement is permitted by these universal joints, which have proved highly satisfactory in service. They are relatively inexpensive to manufacture, deal effectively with the angularities involved, and require no maintenance.

The outboard yoke of the joint forms part of the sliding splined coupling necessitated by the use of front suspension of the double transverse link type. Consequently, it has a longer hub than has the inboard yoke, and the inboard end of its bore is sealed by a domed cap, pressed into a shallow counterbore and sealed by peening. A convoluted gaiter of synthetic rubber is secured by clips to the outboard end of the yoke and to the half-shaft. This gaiter serves as a seal for the molybdenum disulphide treated grease with which the coupling is packed on assembly, and it has sufficient flexibility to accommodate not only the relative axial movement of the components but also the changes of volume that this movement produces. Each member of the sliding coupling has 16 splines of involute form, and the overall diameter of the splines on the half-shaft is 1 in.

The nominal distance between the geometric centres of the universal joints of the left-hand half-shaft is less than that for the right-hand shaft, because the final-drive unit is situated to the left of the longitudinal axis of the car. Whereas the left-hand dimension is $13\frac{1}{16}$ in, that on the right is $16\frac{1}{16}$ in. For most of their length, the half-shafts have a nominal diameter of $\frac{7}{8}$ in. At the outboard end of each is a Birfield type 75AC constant-velocity universal joint.



Ballbearings, clamped between the sump and the final-drive casing, carry the large gear and differential cage

This joint was developed specially for the ADO 15 vehicle and is the smallest of its type at present produced by Hardy Spicer Ltd. Although the design of the joint is based on the original Rzeppa principle, it has been improved in several respects and is now covered by British patents. The 75AC joint is noteworthy for its small overall diameter —only 3 in—in relation to its static torque capacity of 6,820 lb-in, and for the small number of component parts. These comprise an inner member, a ball cage, six balls and an outer member; the inner member is mounted on splines on the half-shaft and the outer one is integral with the stub axle for the wheel hub, the drive being transmitted from one to the other by means of the balls, which are positioned in tracks in the members.

The inner and outer members and the cage are all forgings of case-hardening steel, the material of the two members being of the nickel-chromium alloy type. Standard bearing balls of $\frac{9}{16}$ in diameter are employed. Over most of its length, the stub axle portion of the outer member has a nominal diameter of $\frac{7}{8}$ in, and on it are machined 19 splines of involute form. Its extremity is of $\frac{5}{8}$ in diameter and is threaded to take the nut securing the hub assembly.

As is well-known, the constant-velocity characteristic is only obtained if the plane of the intermediate member of a joint bisects the angle between the driving and driven shafts. In the Birfield joint, of course, the intermediate member consists of the balls, which are restrained in the bisecting plane by the combination of the cage and the configuration of the tracks in the inner and outer members. The outer surface of the inner member and the inner one of the outer member are ground to a spherical form having a common centre when the joint is assembled. On both members, the ball tracks are ground to radii, the centres of which are offset axially from the common centre. The offset is towards the open end of the outer member, and in the other direction on the inner member, so the result is an opposing convergence of the two sets of tracks.

Both surfaces of the cage are spherically ground, and the component is a close running fit on the inner member and within the outer one. Because of the convergence of the tracks, the windows in the cage are circumferentially elongated, but have only a nominal axial clearance over the balls. Two of the windows, one at each end of a common diameter, are further elongated, to permit assembly of the joint without the use of tools.

The configuration of the tracks is such that the sum of the depths of the two holding one ball is constant, in the bisecting plane, for all angular deflections of the joint up to the maximum of 40 deg. Because of the opposed offsets of the centres of the track radii, however, the balls tend to move axially in the direction of greater clearance; since this direction is the same for all the balls, such movement is resisted by the cage, which therefore restrains them in the bisecting plane.

This centring action is facilitated by the section of the ball tracks which, instead of being a circular arc, is part-elliptical. The form of the ellipse is such as to produce a pressure angle of 45 deg. It follows that the balls are in compression instead of shear, the lines of contact being well away from the edges of the tracks. The balls are enabled to roll freely at all joint angles because the conformity—that is, the ratio

In the Birfield type 75AC constant-velocity universal joint, six balls transmit the drive from one member to the other. The joint is very compact and is able to accommodate a maximum articulation of 40 degrees

of the instantaneous radius of curvature, at the point of contact, to the radius of the ball—is approximately 1·04:1.

The half-shaft is a push fit in the inner member of the joint, which is retained against a shoulder by means of a spring ring housed in a groove round the shaft; a chamfer at the end of the bore of the member forms a register for the ring. Because assembly of the joint is only possible prior to the insertion of the shaft, and the shaft end is inaccessible after installation, the ring is fitted into the groove before the shaft is offered up. The groove depth is sufficient to accommodate the full section of the ring during the entry of the shaft. As full engagement is reached, the chamfer comes into the plane of the groove so the ring springs outward automatically to secure the member. This simple method of fixing has proved capable of withstanding axial loadings well in excess of those encountered in the severest driving conditions.

The joint is lubricated by grease, with which it is packed on assembly. A convoluted synthetic rubber gaiter is subsequently installed to retain the grease, which does not require replenishment during the normal working life. One end of the gaiter seats in a shallow peripheral groove on the outer member, and the other in a similar groove on the half-shaft. Both ends are secured by aluminium rings that are slipped on and then crimped to grip the rubber.

SPECIFICATION

ENGINE: Mounted transversely in car. Four cylinders. Bore and stroke, 62·9 mm and 68·26 mm. Swept volume, 848 cm³. Compression ratio, 8·3 : 1. Maximum gross power output, 37 b.h.p. at 5,500 r.p.m. Maximum gross b.m.e.p. 132 lb/in² at 2,900 r.p.m. Maximum gross torque, 44 lb-ft at 2,900 r.p.m. Three-bearing, counterbalanced crankshaft. Chain driven, side mounted camshaft. Vertical, in-line overhead valves, actuated by push rods and pressed steel rockers. Heart shape combustion chambers in cast iron cylinder head. Siamesed pairs of inlet ports; middle pair of exhaust ports siamesed. Single iron casting for inlet and exhaust manifolds. S.U. HS2 carburettor, with 1¼ in diameter throttle barrel. Paper element air filter. Electric fuel pump. Dry weight, complete with radiator and fan, clutch, gearbox and final drive assembly, and electrical equipment, approximately 308 lb.

TRANSMISSION: Single-dry-plate clutch, 7⅛ in diameter, with hydraulic actuation. Four-speed gearbox assembly housed in sump and driven from clutch output member by train of helical gears. Synchromesh on top, third and second gears; floor type manual control. Ratios: top, 1 : 1; third, 1·412 : 1; second, 2·172 : 1; first and reverse, 3·628 : 1. Final-drive reduction of 3·76 : 1, by helical spur gears; pinion mounted on gearbox mainshaft. Two-pinion differential. Drive to front wheels by swinging half-shafts, each incorporating splined sliding coupling and rubber bushed universal joint at inboard end; outboard universal joints of constant-velocity type. Speed per 1,000 r.p.m. in top gear, 14·85 m.p.h.

Special Slide Rule

SLIDE rule type calculators designed for special applications are being used increasingly widely throughout science and industry. This has been brought about by close co-operation between potential users and manufacturers of slide rules. One of those recently introduced is for the calculation of pipe weights and strengths. It has been designed by Mr. I. Alexander, of Intercontinental Enterprises, and is produced in the Regulus Works of Blundell Rules Ltd, at Lynch Lane, Weymouth, Dorset. The purpose of the slide rule is to facilitate the calculation of solutions to problems that frequently recur.

The instrument gives the answers, in both U.K. and metric units, to three commonly recurring sets of calculations, which otherwise would take a considerable time to work out. One operation gives the weight of steel, aluminium, cast or spun iron, bronze, copper or lead pipes when the diameter and thickness are known. Another operation enables the user to determine the value of either: bursting, test or working pressure; the yield or ultimate stress of material required; or the outside diameter or thickness of a metal pipe when the other three values are known. Finally, there are scales to give the relative carrying capacities of pipes of different internal diameters. It is also possible to use the rule for straightforward price calculations.

Electronic Aids for Industry

AT THE 6th International Instrument Show, to be held in London from 19th to 23rd June 1961, over 50 exhibitors drawn from 10 countries will be showing the most up to date electronic instrumentation available. The exhibition is expected to be even wider in scope than in former years, and much of the equipment on view will be seen in this country for the first time. Those partaking will come from Canada, Denmark, France, Holland, Italy, Sweden, Switzerland, U.S.A. and West Germany as well as Great Britain, and will show instrumentation covering such varied fields as acoustic measurement, aircraft maintenance, nuclear analysis, radar, automation and quality control. As in previous years, the event will provide a valuable opportunity for scientists and engineers working on research, development and production tests for a diversity of British industries to see in operation the latest range of foreign electronic instrumentation prior to an actual purchase being made. The show is organized by B and K Laboratories Ltd, whose premises are in Park Lane, London, W.1.

Engineering Equipment Users' Association

A FIVE-PAGE booklet has been issued by the Engineering Equipment Users' Association: it is entitled "A Short Description of the Association's Objects and Organization". This Association, which was incorporated in 1950, aims at providing an effective means for co-ordinating user information, and for the presentation of that information alongside the views held by manufacturing interests. This booklet can be obtained from the Association, whose address is 20 Grosvenor Gardens, London, S.W.1.

Welding of Aluminium

SINCE the publication, in 1955, of The Aluminium Development Association's Information Bulletin No. 19, entitled "The Arc Welding of Aluminium", considerable progress has been made with the development of the two inert gas shielded arc processes now known as TIG — inert gas, tungsten arc—and MIG—inert gas, metal arc. A supplementary publication has now been issued: it is entitled "Notes on Process Selection and Filler/Electrode Materials". This new publication also covers the developments of fine wire welding with the MIG gun and amplifies the paragraphs of Bulletin 19 that deal with the choice of filler rods or electrodes to suit the welding of certain parent metals.

The booklet is divided into two parts. The first gives further guidance on the choice of welding processes, and the second deals with the appropriate fillers or electrodes for welding pure aluminium to one of its alloys or dissimilar metals. It also covers the welding of wrought to cast aluminium. The address of The Aluminium Development Association is 33 Grosvenor Street, London, W.1.

"Foolproof" Tape-Controlled Milling

New EMI System Prevents the Execution of Erroneous Parts of Programmes

A DEMONSTRATION was recently given in London of a new development in tape-controlled milling as applied to operations such as die-sinking. EMI Electronics Ltd, of Hayes, Middlesex, were showing their "EMICON" punched-tape system in which a safety device is incorporated which detects any programme errors and eliminates inadvertent departures from the planned profiles. Hitherto, tape-control systems have been designed to stop the machine as soon as a linear error has been made; the drawback in such systems, of course, is that the error might well be of sufficient magnitude to damage the workpiece beyond repair. This can be both time-wasting and expensive, particularly if it occurs towards the end of a complicated machining programme. In the EMICON system a "Safety Angle" continuous checking device detects an error before incorrect information can be passed on, and the machine is automatically stopped to enable the fault to be remedied.

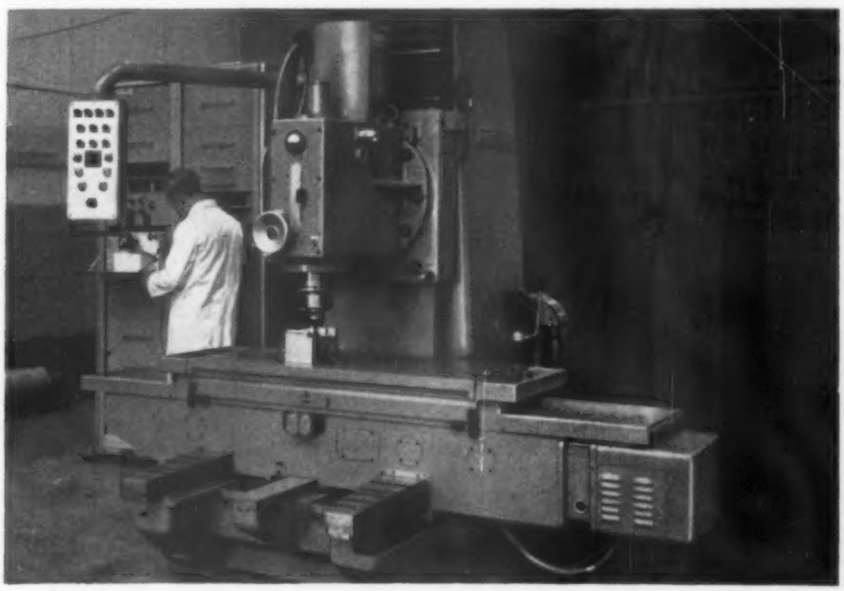
There are three phases in the EMICON control system: reading, storing and interpolating. The punched tape that is fed into the reader contains information in the form of co-ordinate dimensions for either full three-dimensional machining, or stepless contours in two axes and very fine step control in the third. The machine controls translate these into lateral movements of the table and cross-slide, and vertical movements of the milling head. This information is fed in small batches into the store so that, in turn, it is able to maintain a steady flow of information to the interpolator while the machine is obeying previously transmitted instructions.

It is the interpolator that embodies the "Safety Angle" device for detecting errors. The co-ordinate points are scanned in groups of three, the last point of one group also being the first of the next group, and a parabolic curve is interpolated so that these points blend into a smooth contour. If blending is found to be smooth, the information is passed on to the machine's control system, but if the interpolator finds that the contour is attempting to deviate from a blended line, then the safety device transmits a signal that switches off the machine.

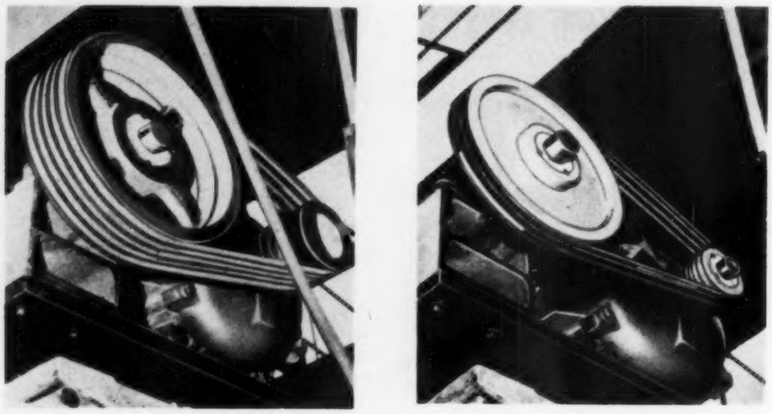
The angle by which the faulty line deviates from a blended contour is the deciding factor in this system, hence the title "Safety Angle". This angle can, of course, be chosen to suit surface finish, depth of cut and material being worked. In the programming stage it is not necessary to quote every co-ordinate point, but merely suitable blend points, since the interpolator will supply the rest. However, if precise co-ordinate points are fed into the programme, the interpolator will scan them and act in the capacity of a checker.

A valuable feature in the EMICON system is that the input programme can be corrected for the actual size of the cutting tool, even after resharpening has been performed during the course of a cutting operation. This system also allows for zero shift in all axes, so that accurate positioning of the workpiece is effected and setting up time is reduced to a minimum. In the event of an intentional abrupt deviation being included in the machining programme, special coded instructions can be embodied in the programme tape, so that the safety angle unit is temporarily by-passed.

The machine used in this demonstration is the type FS 80 N vertical mill of the Droop and Rein Company, of Bielefeld, Germany, and is a modified version of the firm's standard copy-milling machine. Its table size is 61 in × 21 in, and the range of movements of the table, cross-slide and headstock are 39 in, 17·5 in and 19·5 in respectively. In Great Britain, the agents for the Droop and Rein range of equipment are Elgar Machine Tool Co. Ltd, whose address is 172-178 Victoria Road, Acton, London, W.3.



A Droop and Rein vertical milling machine equipped with the EMICON punched-tape control system. This incorporates a "Safety Angle" unit for stopping the machine before faulty information can damage the workpiece beyond repair



These two illustrations effectively show the saving in width that can be obtained by the use of a SpacesaVer drive in place of one of the normal type. In this drive, five standard belts have been replaced by three of the narrower SpacesaVer belts

INDUSTRIAL BELT DRIVE

Notes on the Fenner SpacesaVer Design, which Substantially Reduces the Size, Weight and Cost of Installations

ALTHOUGH the normal V-belt drive is both efficient and reliable, as is witnessed by its widespread use, it is liable to be heavier and more bulky than certain other driving systems. These disadvantages appear to be overcome, however, in the SpacesaVer wedge-belt drives, an improved range recently introduced, for machine tools and other industrial purposes, by J. H. Fenner and Co. Ltd, of Hull. The design of the belt employed on these drives is not new: in fact, considerable experience has already been gained with it in single- and two-strand applications on motor vehicles, driving fans and other ancillaries. These automobile drives were, of course, specially tailored for their particular duties, whereas the SpacesaVer series is of the standardized type, covering the full range from 1 to 200 h.p.

For a given transmission capacity, the face width of a SpacesaVer drive is considerably narrower than that of an orthodox V-belt drive. In general, the reduction in width is in the region of 50 per cent, but higher figures have been achieved in some applications. This saving results from the fact that the belts not only are of narrower section than their predecessors, but also have a considerably higher rating, so the number necessary for a given multi-strand application can be reduced. The more flexible belt section permits a reduction in the pulley diameter; the centre distance, too, can be lessened, without detriment in respect of the durability of the drive.

Because of these features, the minimum volume that a SpacesaVer drive can occupy is considerably less than that of the equivalent normal drive, and the weight is likely to be some 40 per cent lower. From the reliability aspect, there will obviously be an improvement in shaft bearing life as a result of the reduced pulley overhang. It also noteworthy that the cost of a SpacesaVer drive should be from 20 to 35 per cent less than that of previous designs.

It has been found that these drives can be operated satisfactorily at higher speeds than can those of earlier type, an obvious advantage in many applications. Belt linear speeds of up to 6,000 ft/min are now practicable, whereas the previous limit was in the region of 5,000 ft/min. For the range of powers mentioned earlier, only two sizes of belt—$\frac{3}{8}$ in and $\frac{5}{8}$ in overall width—are necessary, with a resultant simplification of production. To cater for the transmission of still higher powers, however, a belt of 1 in overall width is to be added to the range in due course.

The ability of the new type of belt to transmit a greater pull than a normal belt of larger section is attributable to several factors. First, as a result of intensive research, the shape of the section has been altered: it is narrower than before, in relation to the depth. Because of the reduced overall width, there is less tendency for the belt to buckle, and so for the cords to move and generate internal heat. This greater stability of the cords is markedly assisted by the concave form adopted for the base, and by the slight convexity of the top, or widest, face of the belt.

Materials of higher grade than hitherto are employed. The cords are of Terylene, the strength and flexibility of which are substantially greater than those of the equivalent cotton or rayon cords. An earlier problem in the use of polyester cords, such as Terylene, was that of bonding them permanently to the synthetic rubber of the core, which has an olefinic structure. According to J. H. Fenner and Co. Ltd, this difficulty has now been completely overcome. The belt has adequate electrical conductivity to deal with normal static effects.

Because of the careful production control and the accuracy with which the length of these belts can be measured in the factory, selective matching for a multi-strand drive can be carried out with precision. Moreover, the consistency of the materials used ensures that, within very narrow limits, all the belts of a matched set take an equal share of the load. Consequently, the belts have very little tendency to wear unevenly in service.

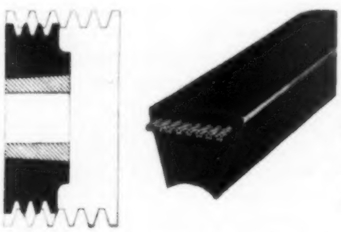
Drive comparisons and selection

The accompanying table, which covers four typical drives, indicates the savings in size, weight and cost that are possible by the use of SpacesaVer drives in place of conventional systems. These drives range from 5 to 100 h.p, and in each case the main particulars of directly comparable V-belt and wedge-belt types are quoted, together with the percentage reductions effected in the relevant figures. All the drives are included in the company's standard lists, so exact prices can be quoted.

The range of standard drives is comprehensive. In the Alpha series, which employs the $\frac{3}{8}$ in wide belt, it covers pulleys having from one to six grooves and providing ratios

Table—COMPARISON OF DIMENSIONS AND COST OF V-BELT AND WEDGE-BELT DRIVES IN FOUR TYPICAL APPLICATIONS

	5 h.p.			25 h.p.			50 h.p.			100 h.p.		
	V-belt	**Wedge-belt**	**Saving per cent**	**V-belt**	**Wedge-belt**	**Saving per cent**	**V-belt**	**Wedge-belt**	**Saving per cent**	**V-belt**	**Wedge-belt**	**Saving per cent**
Number and type of belts employed	five A29	three Alpha 280	—	six B65	six Alpha 560	—	six C112	five Beta 1000	—	eight C180	six Beta 1600	—
Motor pulley dia—inches	3¾	3·15	16	6½	5·3	18	8¾	7·1	19	10¾	9·0	16
Driven pulley dia—inches	6⅜	5·6	12	12½	10·6	15	24¾	21·2	14	42¾	37·5	12
Face width—inches	3¼	1½	54	5 7/16	2 13/16	48	6 13/16	3¾	45	8 13/16	4 7/16	50
Centre distance—inches	7·5	7·1	5	18·7	15·3	18	30·5	26·9	12	47·2	41·1	13
Total weight of drive—lb	16·7	10·6	36	66·8	36·2	45	258	138	46	530	340	35
List price	£ s d 6 8 7	£ s d 5 3 8	19	£ s d 17 0 7	£ s d 11 11 6	32	£ s d 45 0 1	£ s d 32 19 9	26	£ s d 86 2 4	£ s d 63 15 2	26



Left: A comparison of the diameters and widths of a standard V-belt pulley and its SpacesaVer equivalent. Right: Terylene cords are a feature of the new belts

of from 1 : 1 up to 7·45 : 1. In the Beta series, the ⅝ in wide belt is used, and the pulleys have from three to six grooves; the maximum reduction is 7·04 : 1. In this connection, it is worthy of mention that the company has produced a comprehensive catalogue, which not only covers every standard drive but also includes selection data both for these and for drives outside the standard range. Copies of this catalogue, reference 135/20, are obtainable from the manufacturers.

Taper-Lock attachment

To facilitate mounting, every standard pulley in the SpacesaVer series is fitted with the Fenner patented Taper-Lock bush. This fixing method is simple and provides a secure grip on the shaft; nevertheless, the pulley can readily be moved if required, for alignment purposes or a change of ratio. The Taper-Lock bush is of high-grade cast iron, and its periphery is taper turned to an included angle of 8 deg. It is accommodated within a similarly tapered bore in the pulley casting.

The bush is bored to be a sliding fit on the shaft, and has a full-length diametral slot to provide the necessary circumferential flexibility. Not only does this slot split the wall of the bush on one side of the axis, but it also extends from the bore part-way through the wall on the other side. This feature minimizes distortion of the bore from the circular when the bush is pulled into the taper in the pulley, to clamp it on to the shaft. The pulling-in is effected by tightening the hexagon-socket bolts—two or three according to the pulley size—spaced round the periphery of the bush, parallel to its axis.

In the bore is a normal keyway, though the grip on the shaft is normally sufficient for a key to be unnecessary. For pulling the bush out of the taper, and thereby releasing its grip on the shaft, there are one or two additional holes extending through the bush into the pulley; in the bush, each hole is threaded, whereas that portion of it in the pulley is of clearance size, with a blind end. Since the thread size is the same as that of the pulling-in bolts, no additional extractor bolts are required, it being only necessary to screw in the appropriate number of pulling-in bolts after all have been unscrewed from their original positions.

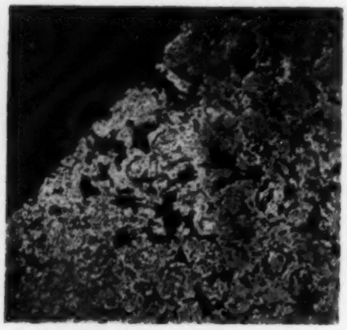
Power Supply Unit

IN LABORATORY work, the requirements in respect of the electricity supply frequently vary widely, so a supply unit with a considerable range can be valuable. Equipment of this type has recently been added to the products manufactured by Pye. It is known as the Stabilized Power Supply Unit, reference number 8440, and operates with an input of 110 to 120 or 190 to 250 volts at 40 to 60 c/sec. Normally it provides a d.c. output which, in the positive direction of current flow, is infinitely variable over four ranges between 0 and 500 volts. In addition, an independent low-voltage a.c. output is obtainable; this, however, is unstabilized, whereas the d.c. output is stabilized to fine limits.

The unit is robustly constructed and its case is well ventilated to permit operation at relatively high temperatures. It is intended for rack mounting, or for building into a permanent installation, but it can be stood on a bench. The overall dimensions are $19 \times 10\frac{1}{2} \times 17\frac{1}{2}$ in, and the weight is 87½ lb. A descriptive leaflet is available from W. G. Pye and Co. Ltd, Granta Works, Cambridge.

Polyester Moulding Compound

INFORMATION has been received from Scott Bader and Co. Ltd. regarding Alpon 5, a polyester dough moulding compound reinforced with sisal fibre. Since the resin is of a tough and fast curing type, high rates of production are possible of any size of moulding within the capacity of the press. In favourable conditions, moulding cycles as short as 30 sec are practicable. The consistency of Alpon 5 is such that the moulding of complicated shapes presents no special difficulty. Among the automobile applications envisaged by Scott Bader are housings for car heater units, interior trim components, and glove boxes. Further details of the material can be obtained from the manufacturers, the address of whom is Wollaston, Wellingborough, Northants.



Photomicrograph of a typical Polyslip 1M bearing, showing how the mixture of p.t.f.e. and lead impregnates the bronze matrix

POLYSLIP 1M DRY BEARINGS

Bound Brook Bearings Ltd. Introduces Improved Range Embodying Polytetrafluoroethylene

TOWARDS the end of 1958, Bound Brook Bearings Ltd, a member of the Birfield Group, announced a range of p.t.f.e-impregnated low-friction dry bearings known as Polyslip. These bearings have now been superseded by the Polyslip 1M series, which represents a considerable technical advance on the earlier type. Not only does the performance of Polyslip 1M bearings exceed that of the earlier pattern by 400 to 500 per cent, but greater standardization of the sizes manufactured has resulted in a reduction in price. The standard sizes are based on B.S.1131, Part 5, 1955: there are 14 bore diameters between $\frac{3}{16}$ in and $2\frac{1}{2}$ in; normally, two bearings of standard lengths are available in each bore size. Special bearings, of both cylindrical and flanged forms, and thrust washers can be produced if the quantities involved are sufficient to justify the tooling costs involved.

In essence, a Polyslip 1M bearing consists of a porous bronze matrix—produced by a sintering process—the working surfaces of which are impregnated with a mixture of p.t.f.e. and lead. The matrix has a composition of 90 per cent copper and 10 per cent tin, and the depth of impregnation is about 0·010 in. In addition, the surfaces are covered by a thin overlay of the same p.t.f.e-base material, the thickness of which is about 0·001 in. The construction of these bearings is covered by British patents 657080, 657085 and others.

Those who read the article on the Glacier DU bearings, published in the May 1960 issue of *Automobile Engineer*, will appreciate the similarity between the physical structures of the Polyslip 1M and the Glacier products. However, the differences between the two are such that there is in fact little overlap between the respective ranges. In the first place, the standard DU bushes are of the thin wall, steel-backed type, whereas the Polyslip 1M components are unbacked and have a greater wall thickness. In addition, the Glacier bearings are wrapped, whereas the others are not; the absence of any discontinuity in the Polyslip 1M bushes can be expected to make them preferable in certain applications.

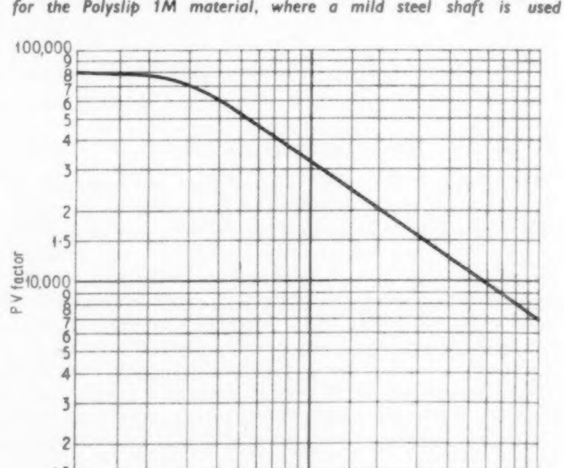
In general, the minimum practicable internal diameter of a wrapped type bush is limited to about $\frac{3}{8}$ in by the process of manufacture. As has already been indicated by the sizes quoted, this disadvantage does not apply in the case of the Bound Brook bearings, which are therefore suitable for use in throttle linkages or other light-duty control systems. The absence of a strip steel backing enables Polyslip bearings to be manufactured, for special purposes, with a spherical external form for self-alignment. Where axial as well as radial loadings have to be accommodated, the bearing can have an integral flange, whereas a separate thrust washer is needed with a DU bush.

Owing to the differences in the construction of the two types of bearing, their mechanical properties—as distinct from their bearing performance—differ appreciably. In impact loading conditions, for example, the structure of the DU type is superior to that of the Polyslip 1M. Finally, it is worthy of mention that there are differences between the respective limiting ratios of length:diameter for the various common bore sizes.

Regarding the performance of the Polyslip bearings, the manufacturers state that the maximum permissible pressure, for low speeds or intermittent use, is 4,000 lb/in^2, this figure being governed by the matrix strength. If a bearing is carrying a mild steel shaft, with the recommended clearances, a PV factor of 6,500 lb/in$^2 \times$ ft/min will result in a life of 1,000 hours in completely dry conditions. The relationship of PV factor and life is shown on the accompanying graph. These figures apply to the case of a rotating shaft and a uni-directional load, where only a proportion of the bearing surface is loaded. The effective PV factor is taken to be 40 per cent higher where the bearing rotates, with a uni-directional load, or where there is a rotating load, as in an out-of-balance assembly. In both these instances, clearly, all the bearing surface is in use.

If a hardened shaft is used, it can be expected that the life will be improved by up to 50 per cent for a given PV figure. At greater PVs, however, the temperature reached by the surface of the shaft may be sufficiently high to reduce its hardness, so the gain will not be as great as that at lower PVs. The bearings perform satisfactorily in contact with shafts of hard anodized aluminium, cast iron and any of the usual steels; the use of stainless steel or a chromium plated surface

Graph showing the relationship between PV factor and running life for the Polyslip 1M material, where a mild steel shaft is used



is advisable in corrosive conditions. Copper-base alloys and very soft shaft materials should be avoided. To attain the full life potential, a surface finish of about 20 micro-in is recommended for the shaft. If the surface is substantially rougher than this, the bearing may have to be derated.

A considerable improvement in performance is obtained if a Polyslip 1M bearing is run in a liquid. This is due partly to the cooling effect and partly to the hydrodynamic reduction in the loading between the bearing and the shaft. It is, of course, necessary that the liquid should not react with the shaft, the bearing matrix or the housing; the p.t.f.e. itself is chemically very inert.

Provided that due allowance is made for the difference between the coefficients of thermal expansion of bronze, 18×10^{-6}/deg C, and steel, 12×10^{-6}/deg C, these bearings can be used over a wide temperature range. The upper limit of about 280 deg C is set by the oxidation resistance of the bronze and the slow chemical change undergone by p.t.f.e. at temperatures appreciably above this figure. At the other end of the scale, the p.t.f.e. retains its excellent frictional qualities at temperatures as low as -200 deg C.

The ability of the bearings to run without any lubrication offers well-known advantages in dirty conditions, as well as in those where the presence of any lubricant could give rise to contamination. In general, however, the Polyslip material is not unduly sensitive to dirt. This is attributable to the relatively soft bearing surface, in which small particles of foreign matter can embed themselves without giving rise to severe scoring of the shaft.

An ultimate tensile strength of 10,000 lb/in² is quoted for the composite Polyslip 1M material. The compressive strength for 0·001 in per in permanent set is 7,500 lb/in², while the ultimate figure is 60,000 lb/in². It is not possible to quote an exact figure for the coefficient of friction, since this can vary widely with the speed, load, clearance and other conditions. In the unlubricated state, the coefficient is likely to be between 0·05 and 0·20, whereas in the lubricated state the probable range is from 0·001 to 0·10. Regarding the effect of speed, since the Polyslip 1M bearings have the freedom from stick-slip that is characteristic of p.t.f.e, they provide smooth operation even at very low speeds. This point was mentioned in connection with both the Glacier DU bearings and the Railko components that were reviewed in the October 1960 issue of *Automobile Engineer*. The actual wear mechanism of the Polyslip 1M bearings is exactly as was described in the article on the DU components.

Cost Control of Plant Services

TO FACILITATE central supervision and cost control, Radiatron, of 7 Sheen Park, Richmond, Surrey, are now offering complete installations for the digital indication and printing of measurements of consumption of materials in plant. Measurements that can be dealt with include integrated flow of fluids, for example, oil, water, gases, steam and compressed air; consumption of electrical power can also be measured by this equipment. The installation is designed to fill the need for automatic plant supervision where the use of more complicated electronic computers is not justified.

This equipment is operated by instruments producing an electrical pulse or closing a switch for each unit consumed. It is based on the Kienzle transistorized digital printer and incorporates counting stores, which retain the information in the event of mains failure. Pulses from the measuring instruments are integrated in the stores. The accumulated pulses in each channel are displayed continuously by electro-mechanical registers or in-line indicators. A programming unit initiates printing at pre-set times for some or all of the channels, and switches are incorporated for each channel, for either totalizing or resetting to zero after printing.

Two types of printer are offered: type D10E registers the channel number and count in a vertical column on 3¼ in wide paper strip at a speed of three lines per second. The second is a carriage type printer, with the designation D10SW, and it is available with carriages of widths up to 620 mm, capable of accommodating 150 characters. It prints one column for each channel, and the time of printing is registered in the first column. The printing speed is approximately two channels per second, and the machine can print up to fourteen digits in parallel and has facilities for instantaneous changeover from black to red. Also, the date and time can be printed, if this is desired.

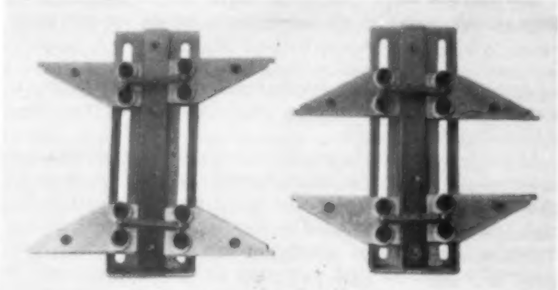
Nickel Plating

AS IS well known, resistance of chromium plated components to deterioration can be improved by using a dual coating of nickel beneath the chromium plate. This type of coating consists of a deposit of semi-bright nickel followed by another of bright nickel. The Electro-Chemical Engineering Co. Ltd, of Sheerwater, Woking, Surrey, have introduced a new process, which they call the Efco-Udylite Bi-Nickel system. It comprises an application of N2E sulphur-free semi-bright nickel, followed by another of 66 bright nickel. It is claimed that this process gives good results even in exposure to marine conditions. Further, it is stated that ductility and freedom from pitting are obtained by virtue of this process, that good levelling and high lustre are obtained and the degree of adhesion between the two deposits is good. Moreover, the overall cost of this system of plating is stated to be relatively low.

Adjustable Gauge

IT HAS been announced by the Speed Tool Co, of 1144N, La Brea Avenue, Los Angeles 38, California, that they are now producing an adjustable go and no-go gauge. This gauge, which can be used for checking either inside or outside diameters, has two pairs of arms, which are independently adjusted and are spring loaded, as shown in the accompanying illustration of the 900 model. The pair of arms on one side, with green identification dots, can be set for a given go dimension, and the pair on the other side, marked with red dots, can be set for the no-go dimension. When the arms are set as shown on the left-hand side of the illustration, the gauge has a capacity of 5 in, and when they are set as on the right, the capacity is 3 in. It is claimed that the accuracy of the gauge has been tested to a millionth of an inch. The price in Great Britain is quoted as $41.50.

The adjustable go and no-go gauge manufactured by the Speed Tool Co.



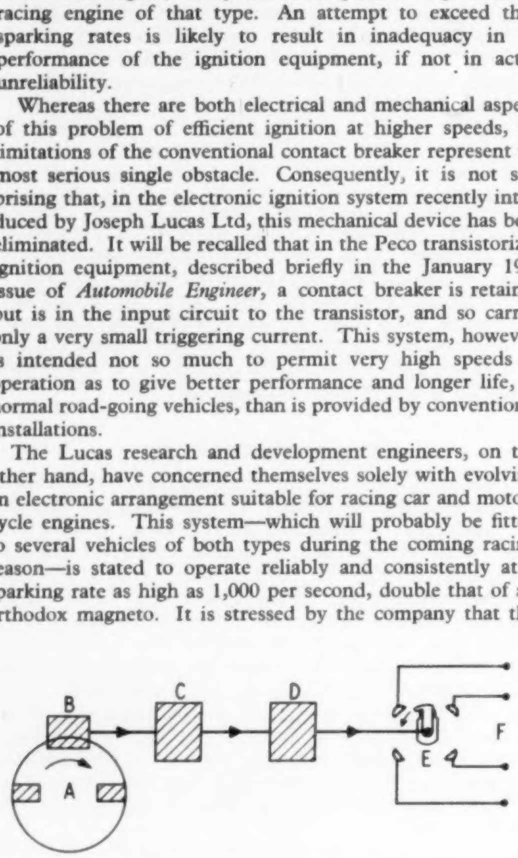
Lucas Electronic Ignition System

Efficiency at Very High Engine Speeds Obtained by the Use of a Transistorized Circuit

SO far as very high-speed engines are concerned, there is little doubt that the limit of development of orthodox ignition systems is being approached. It does not appear possible for existing coil ignition and magneto systems to function effectively at frequencies exceeding about 400 and 500 sparks/sec respectively; the second of these figures is equivalent to 7,500 r.p.m. in the case of an eight-cylinder four-stroke engine, a very moderate speed for a present-day racing engine of that type. An attempt to exceed these sparking rates is likely to result in inadequacy in the performance of the ignition equipment, if not in actual unreliability.

Whereas there are both electrical and mechanical aspects of this problem of efficient ignition at higher speeds, the limitations of the conventional contact breaker represent the most serious single obstacle. Consequently, it is not surprising that, in the electronic ignition system recently introduced by Joseph Lucas Ltd, this mechanical device has been eliminated. It will be recalled that in the Peco transistorized ignition equipment, described briefly in the January 1961 issue of *Automobile Engineer*, a contact breaker is retained but is in the input circuit to the transistor, and so carries only a very small triggering current. This system, however, is intended not so much to permit very high speeds of operation as to give better performance and longer life, in normal road-going vehicles, than is provided by conventional installations.

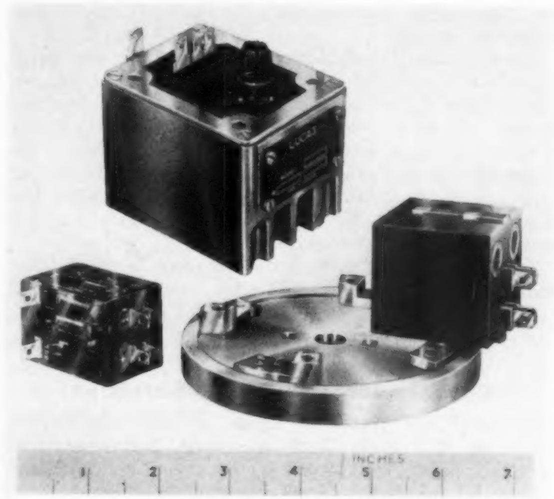
The Lucas research and development engineers, on the other hand, have concerned themselves solely with evolving an electronic arrangement suitable for racing car and motor-cycle engines. This system—which will probably be fitted to several vehicles of both types during the coming racing season—is stated to operate reliably and consistently at a sparking rate as high as 1,000 per second, double that of an orthodox magneto. It is stressed by the company that the equipment is still under development and that, certainly for some time to come, there is no intention of fitting it to other than racing power units.



A *flywheel with pole pieces;* **B** *electromagnetic pick-up;* **C** *trigger amplifier;* **D** *spark generator;* **E** *distributor;* **F** *sparking plugs*

Diagrammatic layout of the sub-assemblies of the ignition system

In brief, the assembly consists of a number of pole pieces—mounted usually on the engine flywheel—an electromagnetic pick-up, a trigger amplifier, a spark generator unit and a high-tension distributor. The flywheel is preferred as the mounting for the pole pieces because it is less affected by torsional oscillations and wear than any other component in the engine. However, for special applications, it would be possible to mount the pole pieces on a light wheel at the other end of the crankshaft. Whichever method is adopted, the pole pieces are accurately positioned, for timing, and their number is that of the requirement in respect of sparks per revolution of the crankshaft. A device for automatic variation of the timing, according to the speed and load, can be embodied if desired; it would seem, though, that on a racing engine having the usual relatively narrow useful speed range, centrifugal control alone would be sufficient, in that it would provide a means of avoiding an excessive amount of advance for starting.



The accompanying scale indicates the size of the units comprising the new Lucas electronic ignition system. Above the flywheel can be seen the spark generator, and the trigger amplifier is on the left

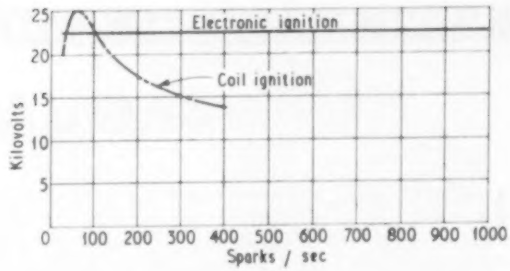
As the crankshaft rotates and one of the pole pieces comes within the field of the pick-up, a voltage impulse is induced in the pick-up and is applied to the trigger amplifier. This amplifier is in effect a switch that is normally closed, and in this condition it permits the flow of current from the battery through the primary winding of a trigger transformer. The applied voltage pulse causes the amplifier to open the circuit, resulting in a cessation of the current flow through the transformer primary winding.

As in an orthodox coil, the collapse of the primary field induces a voltage in the secondary winding of the trigger transformer. This voltage in turn produces a flow of current in the base circuit of the spark generator unit, thereby causing a transistor in this circuit to become conducting. In consequence, battery current is able to flow in the primary winding of a high-voltage transformer in the spark generator. Since the effect of the transistor is to initiate a regenerative oscillation, which results in a very rapid build-up of the primary current, it follows that a high voltage—of over 20 kV—is induced in the secondary winding of the transformer.

This high voltage is fed to the rotor arm of the distributor and thence to the appropriate sparking plug. The rotor arm

is mechanically driven in the conventional manner. Regeneration ceases when the transformer is saturated, and the transistor then again becomes non-conducting; the complete cycle time for regeneration is less than 200 microseconds. When the pole piece moves out of the pick-up field, the trigger amplifier recloses the primary circuit of the trigger transformer in readiness for the succeeding cycle.

Although the spark voltage is independent of the rotational speed of the crankshaft, the current taken from the battery, to operate the system, is proportional to that speed: it rises from about 0·25 amp at 50 sparks/sec to 2·5 amp at 500 sparks/sec. In contrast, of course, a conventional coil ignition system requires more current at low speeds, when the dynamo is probably not charging, than it does at high speeds. The advantage of the electronic system in this respect is obvious.



This graph shows how the voltage of the electronic equipment remains constant up to the maximum rate of 1,000 sparks/sec; in the case of coil ignition, the voltage falls off and the maximum rate is lower

Piezoelectric Strain Gauges

IN the January issue of *Automobile Engineer*, an interesting application of piezoelectric ceramics, in connection with the so-called spark pump, was described. Another use for these materials shows considerable promise, though not yet so fully developed: it is for strain gauges which, of course, are now widely employed in research and development work. Brush Crystal Co. Ltd, of Hythe, Southampton, who manufacture PZT ceramic elements in Great Britain, has already supplied a considerable number of them for use in strain gauge applications. So far, however, the main interest has come from the aircraft gas turbine manufacturers, notably Bristol-Siddeley, de Havilland and Rolls-Royce, though several gauges of this type have been supplied to Aston Martin.

When compared with the orthodox wire type of strain gauge, the piezoelectric unit shows some definite advantages; there are certain minor disadvantages, however, some of which, it is expected, will be overcome as more experience is gained of the appropriate techniques. An obvious point of superiority is that the ceramic element generates the electricity required, so no external source is needed. Since direct readings of current are taken, not only is no balance bridge necessary—thus effecting an additional simplification—but the time required for balancing is saved during a test sequence.

Owing to the high impedance of the ceramic material, slip-ring resistance is much less critical, in the case of tests on a rotating component, than it is where a wire strain gauge is used. In addition, the material has a relatively high permissible maximum stress and is reasonably flexible; the output, which bears a linear relation to the strain, varies little with temperature up to about 150 deg C. A typical element measures perhaps $\frac{1}{2}\times\frac{1}{4}$ in and is 0·010 in thick; it is silver coated on each side, the leads being cemented to the silvered surface. The cost of one of these elements is little greater than that of a wire type gauge.

On the debit side, the material tends to be somewhat over-sensitive to transient shocks, higher readings being given in such circumstances than when the load is applied gradually. Because of its homogeneous nature, PZT ceramic material is prone to what is known as cross-sensitivity where there are two directions of strain mutually at 90 deg; this characteristic, clearly, could affect the accuracy in some applications. Also, a given stress applied in bending results in a different output from that produced by the same stress in tension, but the ratio of the two outputs is consistent.

Another possible source of difficulty is that the response of an element is affected to some extent by the characteristics of the cemented joint to the test piece. For purely practical reasons, it has not yet been found possible to obtain complete consistency in the quality of the joint, which is normally made with an Araldite adhesive. Because of these factors, it is necessary for the strain gauge to be statically calibrated before beginning the test, unless the gauge is being used as a detector rather than a meter.

In the previously mentioned article, reference was made to the converse piezoelectric effect, that is, the ability of the ceramic element to convert an electrical input into a mechanical strain. By virtue of this effect, it is practicable to use a PZT element as a vibrator, for the investigation of resonance problems. However, the applicational possibilities in this direction are limited by the relatively small energy capacity of even quite a large element. It follows that, if large amplitudes of vibration are to be produced, the mass of the system concerned must be low.

Drafting Paper

AFTER intensive research, Ozalid technicians have produced, by entirely new methods, a drafting material that they feel represents a major advance in paper treatment and finishing techniques. Hitherto, the choice of drawing papers has been restricted to two main categories, the treated and the natural papers. Although the natural base has advantages in respect of transparency and quality of surface, it tends to be brittle. Treated papers, on the other hand, are tougher but less transparent and subject to strong discoloration even after relatively short periods of storage: this discoloration reduces the low initial transparency and makes reproduction slow and difficult to achieve satisfactorily.

The new paper is termed Ozavel-D. Examination of a sample has shown it to be tough and to have good tear resistance. It is an all-rag base paper, with a high degree of transparency and white in colour. The manufacturers claim that the material retains its colour even after years of storage. A treatment is applied to both sides of a transparent base to produce durable surfaces that are particularly suitable for either pencil or ink work. Other claims made for the paper are that it will withstand repeated pencil or ink erasures without damage or leaving a ghost image; the lines will not wear or crack from the surface of the paper; because of the fine texture of the surfaces, a uniform density of pencil line can be readily obtained and ink lines neither feather nor thicken on application. The new material is available in rolls or cut sheets, plain or pre-printed. Further details can be obtained from Ozalid Co. Ltd, the address of whom is Langston Road, Loughton, Essex.

NEW BRITISH CARS

JAGUAR E TYPE

BECAUSE of the high reputation of Jaguar cars—achieved mainly by a remarkable combination of excellent engineering, distinctive appearance and value for money—the E type *gran turismo* two-seat model, which was first shown to the public at Geneva last month, has aroused considerable interest. Although this new car, available in open and coupé versions, inevitably has a more limited appeal than others in the range, it is of great technical merit. In brief, it is a compact road vehicle of very high performance—obtained by virtue of a power output of 265 b.h.p. and a low-drag body of small frontal area. Although the car has a maximum speed of about 150 m.p.h, and from a standing-start can cover a quarter of a mile in under 15 sec, it is very tractable, and is finished to the usual Jaguar standards.

With the exception of the rear suspension, which is of the independent layout, the design resembles that of the successful D type, though the body shape has been altered to suit its different function. The makers quote the aerodynamic drags of the D type car and the E type coupé as 151 lb and 184 lb respectively at 100 m.p.h. Since the appropriate frontal areas are in the ratio of 1 : 1·195, it follows that the drag coefficient of the E type car is only about 2 per cent higher than that of the competition model.

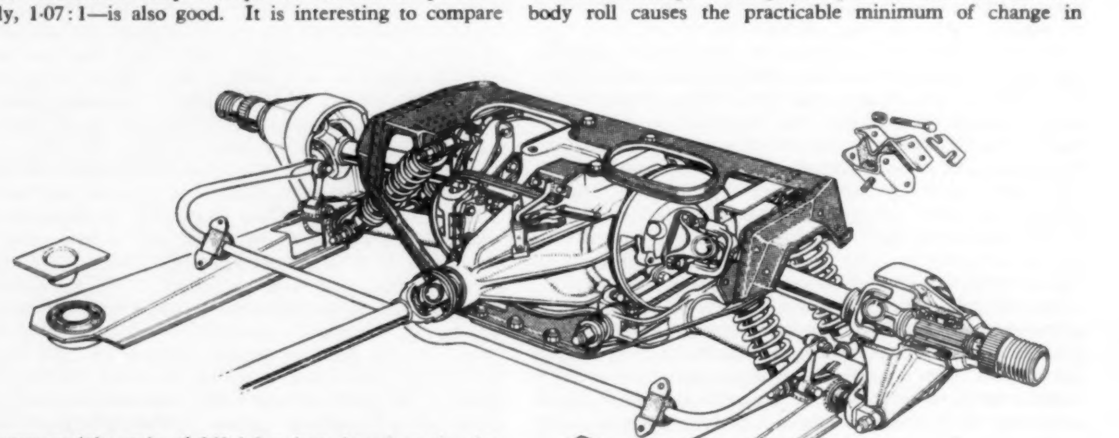
A description of the now familiar 3·8 litre XK Series S engine is unnecessary, but certain aspects of its performance are worthy of comment. At the peak r.p.m. of 5,500, the mean piston speed of this relatively long-stroke unit is 3,830 ft/min, but the maximum piston acceleration is moderate, at 73,300 ft/sec^2. Although the specific output and b.m.e.p. figures of 67½ b.h.p./litre and 172 lb/in^2 are impressive, they are obtained with a compression ratio of 9 : 1, and valve lifts of only ⅜ in; the valve timing, too, is far from extreme: the opening and closing points are 15 deg each side of t.d.c. and 57 deg each side of b.d.c. The 0·727 : 1 ratio for speed at maximum torque : speed at maximum power is satisfactorily low for an engine of this type, and that of maximum torque : torque at maximum power—namely, 1·07 : 1—is also good. It is interesting to compare the power : weight ratio of 241 b.h.p./ton, based on the dry weight, with those of six other *gran turismo* cars: Aston Martin DB4GT, 265 b.h.p/ton Mercedes-Benz 300SL, 183 b.h.p/ton; Ferrari 250GT, 191 b.h.p/ton; Maserati 3500GT, 196 b.h.p/ton; Maserati 5000GT, 263 b.h.p/ton; Facel Vega HK.500, 225 b.h.p/ton.

The double-wishbone front suspension is similar to that of the D type vehicles, and is remarkable chiefly for the fact that the torsion bar springs, since their anchorages to the lower wishbones are inboard of the pivot axes, are also subjected to bending. Though the rear suspension is generally similar to that employed on the experimental car entered privately at Le Mans last year, there are several major differences. The most important of these is that the complete final-drive and suspension assembly is mounted on a fabricated steel sub-frame, instead of on the body structure.

In effect, the sub-frame is a transverse, inverted channel beam, bolted within which is the Salisbury final-drive unit. The suspension is of the double transverse link type, in which the fixed-length half-shafts form the upper links. Each lower link consists of a sturdy tube with a forged fork member at each end; since the links react part of the longitudinal dynamic loadings, the arms of each fork, and the pivot bearings, are widely spaced.

The links, which are longer than the half-shafts, pivot on the sub-frame and on the wheel hub carriers. Each inboard pivot bearing is a double needle roller unit, and the outboard bearings are of the taper roller type. At each side there are two combined coil spring and damper suspension units, anchored to the lower links and to the sub-frame. Reacting the remainder of the longitudinal loading is a pair of pressed steel trailing arms, the rear ends of which house large-diameter rubber pivot bushes of the eccentric type; the pivot bolts are carried in lugs on the lower links. At their leading ends, the arms are attached to the body structure by means of conical rubber mountings having a lower resistance to longitudinal than to lateral movement. An anti-roll bar connects the rear ends of the trailing arms.

It will be appreciated that, because of the rigid pivot bearings of the lower links, the trailing arms do not give rise to any significant roll-steer effect. However, their effective length varies appreciably with suspension movement, and this is the reason for their unusually resilient rubber mountings. The geometry of the links is such that body roll causes the practicable minimum of change in



In the independent rear suspension system, the lower transverse links pivot on widely spaced roller bearings. The trailing arms have rubber mountings, and the final-drive assembly is bolted to a sub-frame

The coupé version of the Jaguar E type. In this view, the low overall height and resemblance to the D type are apparent. The engine is the XK Series S unit, with a power output of 265 b.h.p.

camber angle and track. It is stated that the roll centre height is 5 1/16 in, the spring rate at the wheel is 245 lb/in and the periodicity is 90 c/min. The corresponding figures for the front suspension are 5¼ in, 320 lb/in and 85 c/min.

At each side, the body structure is attached to the sub-frame by two rubber mountings, of the bonded vee type, in tandem. As viewed in plan, these mountings are situated longitudinally, but in side elevation they are at opposing angles of 45 deg to the vertical. As can be seen from the accompanying illustration, vertical and transverse loads are taken by the rubber in combined compression and shear, whereas torsional loads are taken only in shear. In this way, a degree of controlled rotation of the sub-frame and final-drive casing, under positive or negative torque loads, is deliberately permitted, its extent being comparable with that of a conventional live axle mounted on leaf springs. The unsprung weight of this suspension system is said to be less than half that of the Mark 2 cars' live axle layout.

To minimize unsprung weight, the rear disc brakes are mounted inboard, on the output shafts of the differential. Because the heat input to the final-drive unit is greater than if outboard brakes were employed, the outboard bearings of the output shafts have silicone rubber seals. A praiseworthy safety feature of the braking system is that the front and rear hydraulic circuits are independent, each having its own master cylinder. To reduce pedal effort to a comfortable level, a Dunlop vacuum servo unit is incorporated; it is of the type described in the London Show Review issue of *Automobile Engineer,* though the method of application is slightly different.

Although the Jaguar D type has a considerably higher performance than the XK150S models, it is also substantially lighter, so the brake discs are of 11 in diameter at the front and 10 in at the rear, as against the 12 in discs on all four wheels of the XK150S. Since the total swept areas of the brakes of the two cars are 461 in^2 and 540 in^2 respectively, the corresponding figures for swept area per unit of vehicle weight are 418 in^2/ton and 384 in^2/ton, the difference of approximately 9 per cent being in favour of the E type.

The construction of the body is similar to that of the D type cars. A chassisless, stressed skin structure—mainly of 20 s.w.g. steel—extends from the dash to the tail, and to its leading end is attached a tubular space frame assembly, which carries the engine and front suspension. The two main longitudinal members of the primary structure are the box section body sills, and they are bridged by two box sections: one is the dash unit and the other a transverse member just ahead of the wheels. Square section tubing is used for most of the frontal structure, which consists of two separate, laterally triangulated side girders, bridged at their leading ends by a cross member. Extensions forward of the cross member form the pivots for the hinged bonnet and front wing assembly.

There are three noteworthy features of the electrical installation. Parasitic losses are reduced by the use of an electrically driven cooling fan, controlled by a thermostatic switch in the header tank; three windscreen wiper blades are employed, because of the high width : depth ratio of the screen; and the Lucas petrol pump is of a completely new type, which operates completely immersed in the fuel tank.

Jaguar E Type SPECIFICATION DATA

Engine

Number of cylinders 6
Bore 87 mm
Stroke 106 mm
Swept volume 3,781 cm^3
Compression ratio 8 : 1 or 9 : 1
Maximum b.h.p. (gross) 265 at 5,500 r.p.m. on 9 : 1 compression ratio
Maximum b.m.e.p. 172 lb/in^2 at 4,000 r.p.m.
Maximum torque 260 lb-ft at 4,000 r.p.m.
Crankshaft Seven-bearing, forged steel, counterweighted, statically and dynamically balanced
Cylinder head Light alloy, with shrunk-in valve seats of austenitic iron. Segmental spherical combustion chambers
Valves Inclined at included angle of 70 deg. Operation by twin overhead camshafts and inverted bucket type tappets. Camshafts each carried in four bearings; two-stage chain drive from crankshaft
Carburettors Three S.U. HD8, with 2 in diameter throttle barrels
Fuel pump Lucas 2FP, immersed in tank
Dry weight 499 lb, including flywheel, clutch, carburettors and electrical equipment

Transmission

Clutch Borg and Beck single-dry-plate, 10 in diameter
Gearbox type Four-speed, with synchromesh on second, third and fourth
Gear ratios
fourth 1 : 1
third 1·283 : 1
second 1·86 : 1
first and reverse 3·377 : 1
Dry weight 130¾ lb, including bell housing
Propeller shaft Hardy Spicer open type, with needle roller bearing universal joints

Final-drive unit

Type Salisbury three-quarter floating, with hypoid bevel reduction, embodying limited-slip differential unit. Casing carried on fabricated sub-frame, rubber mounted on body structure. Drive transmitted to wheels by fixed-length half-shafts with needle roller bearing universal joints at both ends
Ratios Standard. 3·31 : 1; optional, 2·93 : 1, 3·07 : 1 or 3·54 : 1
Dry weight 274 lb, complete with sub-frame, half-shafts, wheel hubs and rear suspension assembly

Suspension

Front Double transverse wishbones and torsion bar springs, with Girling telescopic dampers and anti-roll bar
Rear Independent, by means of double transverse links and trailing arms; half-shafts form upper transverse links. Two combined coil spring and Girling telescopic damper units on each side. Anti-roll bar connected to rear ends of radius arms

Steering

Type Rack and pinion
Turning circle 37 ft
Turns from lock to lock 2½
Steering wheel 16 in diameter, aluminium; column adjustment for height and reach

Brakes

Front Dunlop disc type. Discs of 11 in diameter mounted on wheel hubs. Total swept area, 242·24 in^2
Rear Dunlop disc type. Discs of 10 in diameter mounted on output shafts of differential unit. Total swept area, 219 in^2. Separate pads for handbrake
Actuation Hydraulic, with independent circuits for front and rear brakes, and Dunlop bellows type vacuum servo unit acting directly on pedal. Cable operation of handbrake.
Distribution of braking effort
front 60 per cent
rear 40 per cent

Wheels and tyres

Wheel type Wire spoke, with centre-lock attachment. Special wheels available for racing
Tyres Standard, Dunlop RS5 6·40-15 in Dunlop R5 racing tyres optional: front, 6·00-15 in; rear, 6·50-15 in
Pressures, standard tyres
front 23 lb/in^2
rear 25 lb/in^2
Increased by 7 lb/in^2, front, and 10 lb/in^2, rear, if maximum performance is to be used

Construction

Main structure of monocoque type, based on two longitudinal members and two transverse members of box section. Engine and front suspension carried on separate space frame unit, fabricated from steel tubes and bolted to main structure

Dimensions

Wheelbase 8 ft
Track, front and rear 4 ft 2 in
Overall length 14 ft 7 5/16 in
Overall width 5 ft 5¼ in
Overall height 4 ft 0½ in
Ground clearance, laden 5½ in
Kerb weight, with 5 gals fuel
open model 2,520 lb
coupé model 2,576 lb
Weight distribution, at kerb weight
front wheels 50 per cent
rear wheels 50 per cent
Frontal area 16·47 ft^2

Suspension Levelling

Armstrong Hydraulic Device that does not Entail the Use of an Engine Driven Pump

A FEW years ago when, because of its considerable advantages, air suspension camr to the fore, many engineers felt that at least some of these advantages could be obtained much more simply by other means. Among those whose thoughts were directed along alternative lines was Armstrong Patents Ltd, of Fulford, York. The principal claims for air suspension—which therefore have to be achieved by a competitive system if it is to be a success—are: a variable rate, insulation of the sprung mass from shocks and vibrations transmitted from the road, and adjustment of the height of the sprung mass automatically with variations of the load carried.

As regards insulation, both the coil and torsion bar type, especially if rubber mounted, afford reasonable insulation against vibration. Even the semi-elliptic spring, provided it is properly protected and suitable interleaving material is employed, gives acceptable results; moreover, it can be relatively easily arranged to give a variable rate, if desired. This leaves just the problem of automatic adjustment of height to be solved.

Obviously, adjustment could be effected by varying the height of the spring anchorage. Of all the devices that might be employed for this purpose, a hydraulic jack seems to be the simplest, and most reliable and compact. However, these advantages would be lost if it were necessary to use an engine-driven hydraulic pump, but to a firm already producing hydraulic pumps in the form of suspension dampers this did not present an insuperable obstacle. All that was necessary was to modify the dampers, by fitting additional valves, so that they would supply the oil under pressure for the actuation of the jack. This arrangement has the fundamental advantage that the distances between the pump and the other components of the system are small and, therefore, the piping arrangement inexpensive and simple. Moreover, there is no need for an extra connection between the sprung and unsprung masses, to sense the height, since the normal linkage for actuating the dampers can be utilized.

The essentials of the system that has been evolved are simply a lever arm type damper assembly, suitably modified to perform the pumping and automatic levelling functions in addition to the damping of the suspension, a hydraulic jack, interposed between the spring and its anchorage or seat, an oil reservoir and some short hydraulic pipes. Normally, there would be only one reservoir and, of the other components that have been mentioned, one for each spring. If desired, the system can be further refined by the incorporation of a hydraulic accumulator, by virtue of which the height of the sprung mass can be regulated when the vehicle is stationary, otherwise this can be done only when the suspension is moving up and down. The arrangement of these components is shown diagrammatically in Fig. 1. An application to an auxiliary spring is illustrated in Fig. 2, and Fig. 3 shows how the system might be applied, for levelling, when air springs are used. The advantage of the latter arrangement is that the need for a compressor is obviated and pneumatic valves—liable to give trouble owing to freezing of condensation in very cold weather—are dispensed with.

So far as manufacture is concerned, Armstrong Patents Ltd. are particularly well placed for producing the components of this system. The jack is similar in form to the telescopic type of damper, which the firm is already making

Fig. 3, right. It is possible that the hydraulic levelling device could be used in conjunction with a sealed air spring, thus obviating the need for a compressor and pneumatic valves

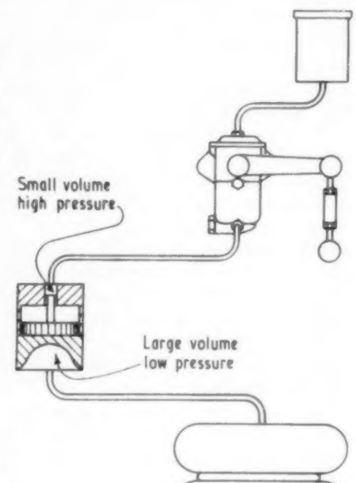
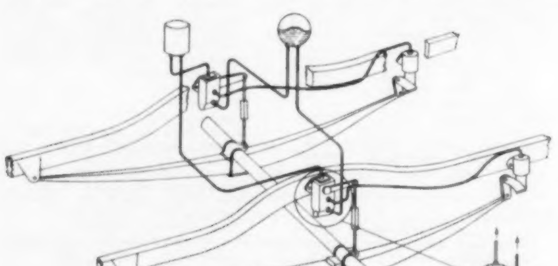
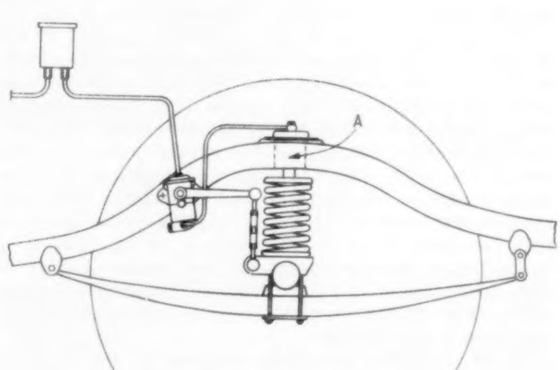


Fig. 2, below. The levelling system applied to an auxiliary spring layout. The hydraulic jack is at A, where it acts on the upper end of the coil type spring

Fig. 1. Diagram showing the layout of the components of the Armstrong hydraulic levelling system with, in the bottom right-hand corner, a scrap view of the pump-damper and its various hydraulic connections





in very large quantities. Basically, the pump-damper unit would be much the same as their existing range of lever arm type dampers, and the other components are very simple.

Modified suspension damper

Two forms of pump-damper unit are illustrated. That in Fig. 4 is for a system in which a hydraulic accumulator is not employed. The outlet to the hydraulic jack is in the base of the unit. Movement of the suspension under the influence of road irregularities actuates the pump-damper unit, which forces oil alternately backwards and forwards through the drilled ducts and the damper valve just below the two cylinders—this of course is the normal damping action. In addition, however, another valve is incorporated: it is of the simple non-return ball type, and is installed with its axis horizontal, near the lower end of one of the cylinders. During the pressure stroke of that cylinder, oil is forced through a radial drilling, past this extra valve, into an approximately vertical duct.

To the lower end of this duct is connected the pipeline to the hydraulic jack, and at its upper end is the height regulation valve, which also is of the ball type. This valve is actuated by a short push rod and a cam plate splined on to the rocker shaft. When the sprung mass is too high, the cam—actuated by the damper arm—depresses the valve: this allows oil from the jack, together with that delivered from the damper, to return to the low-pressure chamber above the two cylinders. If additional load is placed in the vehicle, and the level of the sprung mass falls, the consequent movement of the cam allows the valve to seat, so that the delivery of oil from the shock absorber extends the jack again. Although, with this arrangement, there inevitably is some hunting, this is unimportant because its amplitude and frequency are low. Surprisingly, the performance of the damper is not adversely affected by using it in this way as a pump: this is because, under normal operating conditions, the amount of hydraulic fluid required to effect the levelling is small; in addition, the constant circulation of a small proportion of the fluid helps with regard to cooling and the obviation of aeration. Variations in the quantity of oil required in the jack, as the load changes, are catered for by connecting the reservoir to the low-pressure chamber cover. A fluid whose viscosity does not vary much with temperature is employed, and of course it has additives to increase its film strength, resistance to oxidation and aeration.

The damper illustrated in Fig. 5 is one that has been modified for use in arrangements in which a hydraulic

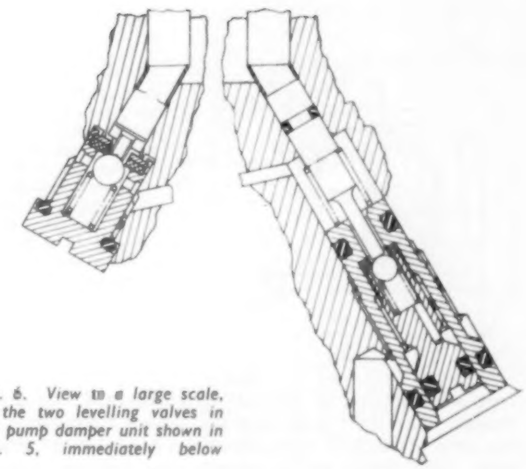
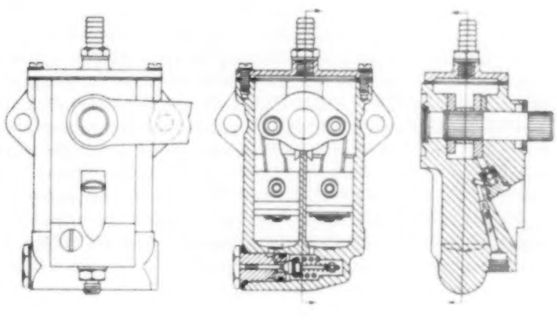


Fig. 6. View to a large scale, of the two levelling valves in the pump damper unit shown in Fig. 5, immediately below

accumulator is employed. In this instance, a non-return valve is installed radially near the lower end of each of the two cylinders. Oil passing these valves is delivered into the vertical passage, as before. However, in this instance, the lower end of the vertical passage is connected by a pipeline to the accumulator, instead of to the jack, and its upper end communicates with a levelling valve for raising the sprung mass. When this valve is opened by the action of the push rod and cam plate, the oil under pressure flows not to the chamber above the two cylinders, as in the unit previously described, but through a transverse duct and restrictor to another almost vertical passage. The lower end of the latter is connected by a pipeline to the hydraulic jack, while its upper end communicates with another levelling valve, also actuated by a cam plate: this latter valve comes into operation only for lowering the sprung mass and it discharges into the low-pressure chamber above the two cylinders. With this arrangement, therefore, there are two cam plates splined on to the rocker shaft, one to actuate the valve for raising and the other that for lowering the sprung mass.



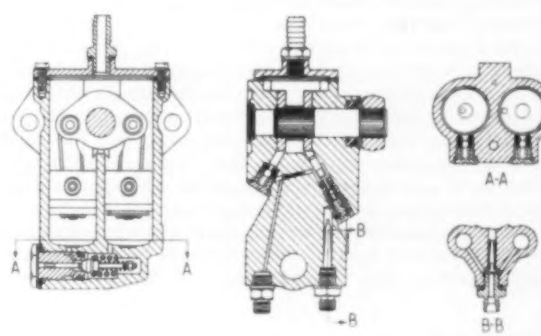


Fig. 4, above left. Three views of the pump-damper unit for application where a hydraulic reservoir is not employed. In this instance there is only one ball type non-return valve through which oil is passed from the lower end of one of the two cylinders into the almost vertical duct; the connection at the lower end of this duct goes directly to the jack, and the valve at the upper end, when opened by the cam and pushrod, allows the oil to return to the low pressure chamber above the damper cylinders. Fig. 5, right. This pump damper unit is for use where a hydraulic reservoir is included in the system. The section AA shows the two delivery valves, one from each cylinder, and section BB the arrangement of the ducts from these valves to the vertical passage. There are two levelling valves, one of them is for raising and the other for lowering the suspension system

Levelling valves

The arrangement of both the simple ball type levelling valves can be seen clearly from Fig. 6. In each, the steel ball seals against a nylon seat which, in the case of the valve for raising the sprung mass, is backed by a shoulder in its thimble

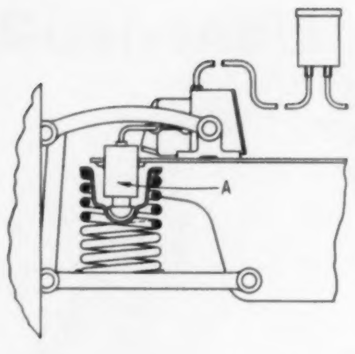


Fig. 11. Diagrammatic illustration showing the basic principles involved in the application of the levelling jack A to the common coil spring and wishbone type suspension

type housing and, in that of the other valve, by a steel washer. The short push rods that unseat these valves and the cams that actuate them are of case hardened, 45 tons/in² steel. Both valves are in thimble type housings, screwed into the zinc alloy die cast body, so that they can be adjusted: this obviates the need for grinding the cams and push rods to close tolerances.

The valve for lowering the sprung mass and that employed in the unit adapted for use without an accumulator are identical: in each case, the push rod is a fairly close fit in its housing, and when the ball is lifted off its seat, the fluid passes up a groove, machined along the push rod, to the low-pressure chamber above the two cylinders. This groove forms a restriction to prevent over sensitivity of the levelling device. As has already been mentioned, in the case of the system incorporating a hydraulic accumulator, a restriction for the valve that raises the level of the sprung mass is fitted in the transverse duct between that valve and the outlet duct to the connection for the jack. All the sealing rings that can be seen in the illustration of the valve are of circular cross section and are of synthetic rubber.

Since the loadings between the faces of the cams and push rods are relatively light, no difficulties have been experienced with regard to wear. The movements of the valves are very small, in the order of 0·040 to 0·050 in. Small movements of the suspension do not have any effect, so far as levelling is concerned, since the cam is inoperative for 1 to 1½ deg on each side of the neutral position. This represents a movement of about ⅛ in, in each direction, of the end of the damper arm.

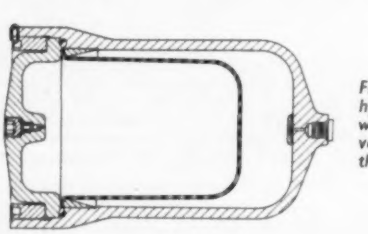


Fig. 7, left. A prototype hydraulic accumulator, with the gas charging valve in the centre of the cover on the left

Hydraulic accumulator

As can be seen from Fig. 7, there is nothing complicated about the hydraulic accumulator, which is a prototype—the production unit of course will be even simpler. In it, the hydraulic oil and the gas are separated by a synthetic rubber diaphragm. The gas compartment is charged with nitrogen at a pressure of 300 to 400 lb/in², according to the installation. This gas is introduced through a ball type non-return valve in the end cover. As a precaution, in case of leakage past the ball valve, a sealing plug is fitted, and the socket is filled so that there is less likelihood of the assembly being tampered with during service. The hydraulic connection at the other end has a shield fitted over it so that the diaphragm, when the hydraulic chamber is empty, will not be damaged by being forced hard against, or even through, the outlet port. Obviously, the size of accumulator will vary with the installation but, in general, it is about 8 in long × 5 in diameter and therefore is not difficult to accommodate, for example, behind a wheel arch.

Hydraulic jack

Illustrated in Fig. 8 is a jack installation for use with a semi-elliptic spring. The arrangement is such that a conventional shackle can be attached to a bracket mounted on the lower end of the ram. An alternative layout would involve the use of the jack as an extendible shackle, by trunnion mounting it at some point between the ends of the cylinder, and by connecting the spring eye directly to the lower end of the ram. However, with this arrangement, a flexible pipe connection is necessary and this is not only expensive but also, since it is in motion all the time the vehicle is in operation, a possible source of failure. The mounting of the jack in the manner illustrated also has the advantage that the height of the space required for its accommodation is less than if the extendible shackle layout is employed.

As can be seen from the illustration, the jack itself is of simple construction. The rod is chromium plated, to minimize the rate of wear, and it operates in a bronze bush. A cast iron piston is employed and it carries a single U-section seal, of a synthetic rubber that remains flexible between the temperatures of −40 deg C and +120 deg C; a hole is drilled from the pressure end of the piston into the groove that houses the seal so that the arms of the U are subject to hydraulic pressure and thus are forced against the bore of the tube and the base of the groove. The bore of the tube is

Fig. 8, below. Typical jack and shackle assembly on a chassis frame, and, on the right-hand side, a cross sectional view of the jack alone

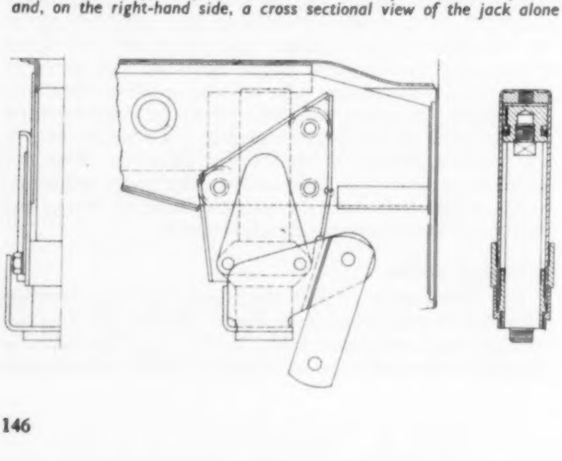
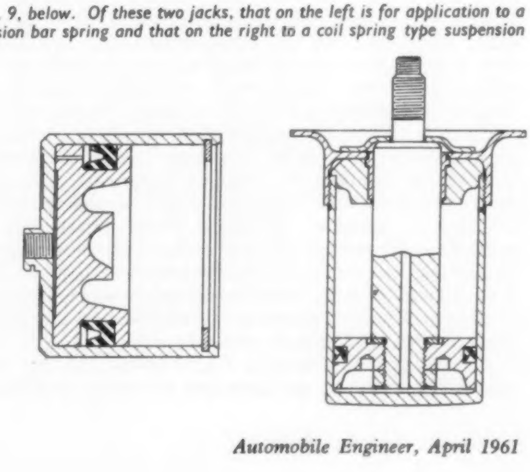


Fig. 9, below. Of these two jacks, that on the left is for application to a torsion bar spring and that on the right to a coil spring type suspension



finished by forcing an oversize hardened steel ball along it. A mild steel end cap is fitted. It has a groove round its periphery in which is fitted a brazing ring. After it has been placed in position, the end of the tube is rolled over to secure it and then induction heated so that the brazing ring melts and forms the seal.

Alternative rams are illustrated in Fig. 9. They are for applications where coil springs or torsion bar springs are employed. Since it is desirable for the pressures in each application to be of the same order, the cross sectional areas of the rams are different. With a semi-elliptic spring, because the ram is applied to only one end, the load on the ram is approximately half that of a comparable coil spring. An even higher loading is necessary with a torsion bar, otherwise an unduly long actuating lever would be required. Diagrams showing suggested arrangements for the levelling devices with suspension systems of these types are shown in Figs. 10 and 11, to the right and left respectively of this column.

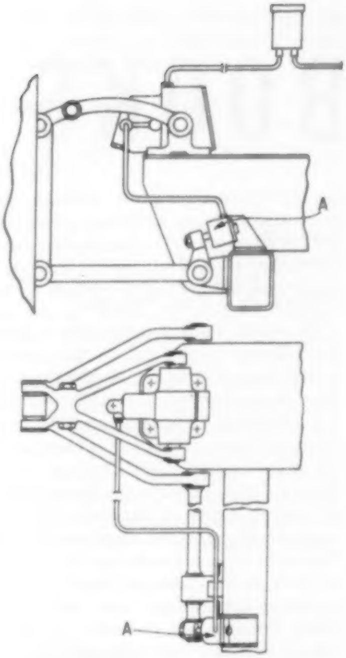


Fig. 10. Plan and elevation of a torsion bar spring suspension assembly, showing how the hydraulic levelling jack A is applied. In this type of application a relatively large diameter jack is required, otherwise the leverage that would be needed would be unacceptably large

General comments

With experimental installations it has been found that it takes from ½ min to 1 min to adjust the level of the sprung mass from the lowest to the normal position, when the vehicle is fully laden. The hydraulic pressure is normally about 300 to 500 lb/in² and the absolute maximum, limited by the setting of the damper blow-off valve, is about 1,400 lb/in². Since the levelling valves seat as soon as the vehicle stops, the ram is then locked hydraulically in the appropriate position. This is an advantage relative to air suspension, with which, if the air pressure falls below a certain level, the suspension springs cannot operate properly until the air pressure has been built up again.

It would appear that the Armstrong system is particularly suitable for application to private cars and light commercial vehicles. This is because of its compactness, low cost, and the fact that additional pumps or other complex components are not required. The system could be designed for application either as standard equipment or as an optional extra. It would probably appeal to motorists who travel during the week without passengers, and fully laden at the weekends. So far as manufacturers are concerned, they could use it either to improve the ride by reducing the spring rate or to reduce the height of the wheel arch. The latter facility would be particularly attractive for commercial vehicle applications. These benefits are obtainable, of course, by virtue of the self-levelling action, which enables the space available for spring deflection to be used always to optimum advantage.

Thin-lead Pencil

LEADS of only 0·020 in diameter are accommodated in the Leroy (R) 020 pencil, which is manufactured by Keuffel and Esser Co, Third and Adams Streets, Hoboken, New Jersey, U.S.A. Because these leads are so thin, they wear down uniformly, so that the lines that are drawn with them are crisp and do not vary in width. Twenty-four leads, in a dispenser, are supplied with the pencil, and extra dispensers are, of course, obtainable.

New Hose

TO SUPERSEDE their two-braid cotton reinforced hose, currently in production, the Goodyear Tyre and Rubber Co. (Great Britain) Ltd, of Wolverhampton, have introduced a long-length single-braid high tensile rayon reinforced hose. The strength of the new hose is equivalent to that of the product it supersedes, the flexibility is greater, while the weight per unit length is less. Among the advantages claimed for the new hose are that it is easy to handle and, since it is available in continuous lengths of approximately 500 ft, the need for couplings is obviated, the possibility of leaks reduced, and there is less likelihood of damage being done to surfaces over which the hose is passed. The rayon is thoroughly impregnated, and there is an adequate safety margin above the specified working pressure. The rubber specification chosen is such as to give good abrasion resistance so that the hose will not be damaged when dragged over rough surfaces. It is a smooth bore hose and is available in ½ in, ⅝ in and ¾ in sizes.

Ergonomics Research Instrument

FOR applications such as the determination of the optimum siting of instruments and controls, and for ascertaining the best possible layouts for machine tools and other equipment, an instrument for the recording of eye movements and lines of sight can be most valuable. Movement of the eye results in the generation of small currents of electricity at the skin surface around their sockets. These currents can be measured, and the movements of the eyes recorded by the means of the Emma equipment developed by E.M.I. Electronics Ltd. The latest version of this equipment is a transistorized model with a self-contained dry battery supply. Its dimensions are only 19 in × 7 in × 9 in, and it weighs 15 lb. Thus it has the advantages of portability and being completely independent of a mains supply. The equipment can be placed on a bench or in any convenient position near to the operator, while the small electrodes are attached to the skin near the subject's eye sockets, and connected by leads to the amplifier of the eye movement detection system.

BOOKS

Brief Comments on Recent Publications Pertinent to Automobile Engineering

Fuel Cells

Edited by G. J. Young

London: CHAPMAN AND HALL LTD, 37 Essex Street, W.C.2. 1960. $9\frac{1}{4} \times 6$. 154 pp. Price 46s.

In recent years considerable interest has been aroused by work being carried out on the development of fuel cells for automotive and other applications. An outcome has been the production of this book, which comprises a series of contributions by leading experts from Great Britain and the United States. The book is based on a symposium sponsored by the Gas and Fuel Division of the American Chemical Society, and it covers the design and principles of operation of various types of fuel cells. In the different sections the contributing authors examine the industrial implications of fuel cells as power supplies for stationary and mobile engines as well as for chemical reactors. Speculative approaches to possible applications have been carefully avoided, and the book provides an accurate, scientifically sound analysis of where and how fuel cells can be applied at present and when and under what conditions they may be employed in the future.

After an Introduction by H. A. Liebhafsky and D. L. Douglas, the contents are as follows: The hydrogen-oxygen (air) fuel cell with carbon electrodes; Catalysis of fuel-cell electrode reactions; Electrode kinetics of low-temperature hydrogen-oxygen fuel cells; The high-pressure hydrogen-oxygen fuel cell; High-temperature fuel cells; Carbonaceous fuel cells; Nature of the electrode processes in fuel gas cells; Molten alkali carbonate cells with gas-diffusion electrodes; Summary of panel discussion; and Index.

The Private Car

London: THE INSTITUTION OF MECHANICAL ENGINEERS, 1 Birdcage Walk, St. James's, S.W.1. 1960. $11 \times 8\frac{3}{4}$. 188 pp. Price 30s.

As is well known by automobile engineers, a series of twelve lectures was presented at meetings of the Automobile Division of The Institution of Mechanical Engineers in the first four months of 1960. This series was given the name The Crompton-Lanchester Lectures, to commemorate two famous pioneers of the motor industry. Each lecture was prepared and read by an eminent automobile engineer, and the whole series reviews recent developments, current practice and probable future trends in car design.

This book is probably one of the most widely useful that it has been our pleasure to review. Since the lectures are presented by leading experts, their technical value is high and they are most useful to practising automobile engineers. Furthermore, one of the aims has been at presenting in simple language the salient features of the art and science of automobile design, and therefore the work can be readily understood not only by engineers in other industries but also by enthusiasts who take a reallv serious interest in automobile engineering as a hobby.

After a Foreword by H. G. Webster entitled "A philosophy of automobile engineering design", the subjects dealt with are: Mechanical design and performance of the petrol engine, by W. V. Appleby; Combustion chamber design and the influence of fuel quality, by D. Downs; Mixture formation, by P.W. Bedale; Automobile transmissions, by J. N. H. Tait; Suspension, steering and tyres, by D. Bastow; Automobile braking, by R. C. Parker; Automobile lubricants and lubrication, by A. Towle; The structure of the automobile, by M. Platt; Electrical equipment of the passenger motor car, by E. A. Watson; Automobile lighting and visibility, by G. Grime; Passenger car comfort, by S. H. Grylls; Performance measurement of the complete car, by A. Fogg; and an Index, giving main headings as well as other details.

Off-The-Road Locomotion

By M. G. Bekker

Ann Arbor: THE UNIVERSITY OF MICHIGAN PRESS. 1960. $9\frac{1}{4} \times 6$. 220 pp. Price $10.00.

In general there is a scarcity of authoritative works on this subject, and therefore the new book by M. G. Bekker is a welcome addition. It is claimed to be the first to offer systematic research on a vehicle as a unit in relation to environmental terrains. The author shows how predictions can be made of vehicle performance and design parameters and discusses soil and snow mechanics, tracks, wheels and tyres, as well as general factors concerning vehicles at assumed optimum performance. He also describes the important developments taking place in industry and education.

The list of contents is as follows: Introduction; Locomotion on wheels and locomotion in nature; The current concept of a motor vehicle and the train concept; Physical properties of soil, mud, and snow; Geometrical properties of terrain surface; The idea of "flotation" and vehicle performance; Soil thrust and locomotion; Motion resistance and vehicular forms; Drawbar-pull and soil-value spectra; Possible improvements and existing trends; The spaced link track; Theoretical concepts and engineering; Operational definition of mechanical mobility; Definition of soil trafficability; Organization and long-range planning of research; Appendix; List of symbols; References; and Index.

The Machining of Steel

By F. C. Lea, O.B.E, D.Sc, M.Inst.C.E, M.I.Mech.E. and Eric N. Simons

London: ODHAMS PRESS LTD, 96 Long Acre, W.C.2. 1960. $8\frac{3}{4} \times 5\frac{1}{2}$. 208 pp. Price 21s.

Fundamental principles of cutting and machining steel and its alloys are presented in this handbook, which is intended for students, apprentices, operators and others interested in engineering production. The work also includes a lot of practical information on the machines and methods used. The only important operation not dealt with is grinding, for which, of course, machines and methods of an entirely different type are employed.

It is the second edition that is under review, and the text has been fully revised and the plan of the book considerably altered. New illustrations have been prepared and much additional matter incorporated, including an account of electro-spark cutting, which is one of the latest developments in the technique of cutting hard metals.

The list of contents is as follows: The lathe; Planing and shaping machines; Single-edge cutting tools; Milling practice; Drilling practice; Broaching practice; Reamers; Sawing practice; Factors in the cutting of steel; Speeds, feeds and coolants; Materials for cutting tools; Jigs, gauges and fixtures; Indexing; Appendix I, Machining manganese steels; Appendix II, Electro-spark cutting; and Index.

Ergonomics of Automation
By A. T. Welford
London: HER MAJESTY'S STATIONERY OFFICE, York House, Kingsway, W.C.2. 1960. 8¾×5¼. 60 pp. Price 3s. 6d.

This book is Number 8 of the series "Problems of Progress in Industry". The object of this series is to present briefly and simply the results of new research into the economic, technical and human problems of industrial progress—those arising from automation and other advanced techniques—and problems of management and social relations. It is also intended to provide a forum for responsible new thinking and to stimulate independent discussion and action, including further research. The conclusions and speculations are those of the investigators, mostly from the universities and other well-known research bodies. Research relevant to the problems of the human operator in an automatic plant, with special emphasis on the design of the equipment, is discussed.

Another booklet, Number 9 in this series, will be published shortly under the title "Automation and Skill". Together, the two booklets constitute the final report of the study, recently sponsored by the Department of Scientific and Industrial Research, of the human implications of automation, as applied in all fields of production and processing.

The contents are as follows: The ergonomic approach to automation; Design for ease of operation; Human capacity for monitoring and control operations; Maintenance work; Some problems of personnel; and Conclusion.

Basics of Fractional Horsepower Motors and Repair
By G. Schweitzer
London: CHAPMAN AND HALL LTD, 37 Essex Street, W.C.2. 1960. 9¼×6. 168 pp. Price 36s.

Fractional horsepower electric motors are, of course, important in many applications, including machine tools and research and development equipment. The purpose of the book under review is to provide a working explanation of the various types of motors of this kind, and to present basic procedures for maintaining them. While the work is primarily intended for those interested in or concerned with the repair of this type of equipment, it is also designed to help the student. For this reason the salient features are presented by means of diagrammatic and other illustrations. The contents are as follows: Basic principles of electric induction motors; Split-phase motors; Capacitor motors; Repulsion motors; Shaded-pole motors; Universal motors; Three-phase motors; Enclosure and mounting characteristics; Windings; Motor testing; Care and maintenance of motors; Control devices; Glossary; Picture credits; and, finally, of course, the Index.

B.M.C. Industrial Gas Turbine

Austin Single-Shaft Unit Recently Announced, Which is Designed to Give an Output of 250 b.h.p.

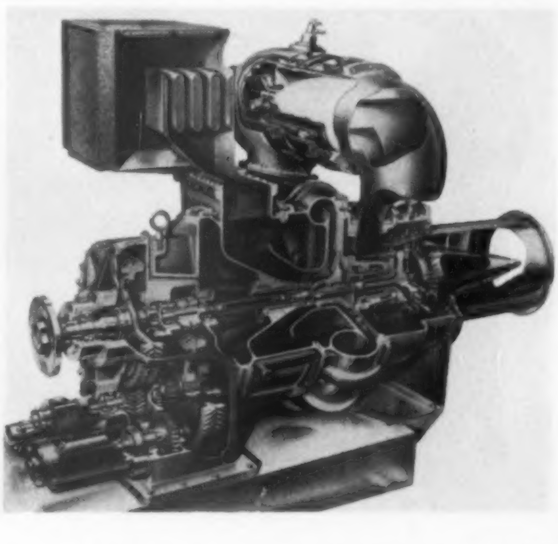
AFTER twelve years of research, the British Motor Corporation has just introduced a 250 b.h.p. industrial gas turbine, designated the Austin 250 unit. According to the manufacturers, this turbine represents a major step in their progress towards the production of a satisfactory automotive version. It will operate on a wide range of cheap fuels and, as would be expected in view of its duty, is of robust construction. At 800 deg C, the cycle temperature is relatively high. Complete with its accessories, the prime mover weighs 900 lb, and its overall dimensions are 37 in long × 35 in wide × 51 in high.

The unit is of the single-shaft type, that is, the rotor of the centrifugal compressor is directly driven by the turbine rotor. Its normal operating speed is 29,000 r.p.m, and at this speed the pressure ratio of the compressor is 3·5:1. To provide an unobstructed efflux, the turbine is at one end of the assembly, adjacent to the compressor. An internally splined hollow shaft couples the common rotor shaft to the input of a double-reduction gear unit at the other end. Between the compressor and the gear case is a plenum chamber from which the compressor draws its pre-filtered air. The compressed air is delivered into a Lucas combustion chamber; fuel is fed to the atomizer in the chamber by a piston type pump. Guide vanes direct the burnt gases into the turbine, which has two stages.

Both reductions are effected by means of double-helical gears, to ensure quiet running and to obviate axial thrust, and the input and output shafts are coaxial. The standard gearing gives an output speed of 1,500 r.p.m, for the generation of electricity at 50 c/sec, but other gearing can be supplied for different applications. No heat exchanger is yet embodied, but a high overall thermal utilization is obtainable in applications where the exhaust heat can be usefully employed. As would be expected, a heat exchanger is under development, and will be introduced at a later date.

If fuel consumption is not of primary importance, the turbine should show considerable advantages over the equivalent diesel engine. Its price is considerably lower, and its compact dimensions and low weight make it easy to transport and install. Starting is fully automatic on pressing one button, and once it is running the unit can be put on full load within a minute. Other benefits are smooth running—bolting down is unnecessary—and low oil consumption, no water cooling system, and a clean exhaust. Little maintenance is said to be required, and the combination of intake and exhaust silencers and sound absorbent covers renders operation at least as quiet as that of a diesel engine.

Part-sectional view of the Austin 250 gas turbine unit. The turbine and compressor rotors are mounted on a common shaft; above them is the single combustion chamber, which was developed by Joseph Lucas Ltd.



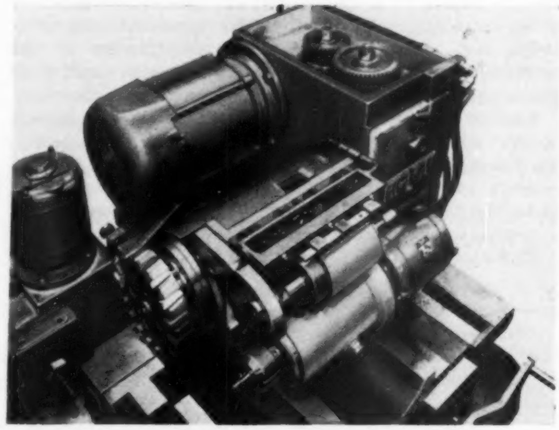
Hi-ton
Facing and Centring Machine

An Automatic-cycling, First-operation Machine for Forged Components

DESIGNED to face and centre drill with rapidity and accuracy both ends of forged components, such as crankshafts and gearshafts, this machine can handle work up to 42 in long and from 1½ in to 6½ in diameter. The minimum length of workpiece that can be accepted is 20 in when two vices are used to clamp the component, but this figure can be lowered to 8½ in if only a single vice is employed. Spindles of the milling heads mount 6 in or 9 in diameter facing cutters and a maximum of ⅜ in stock can be machined from each end of the workpiece.

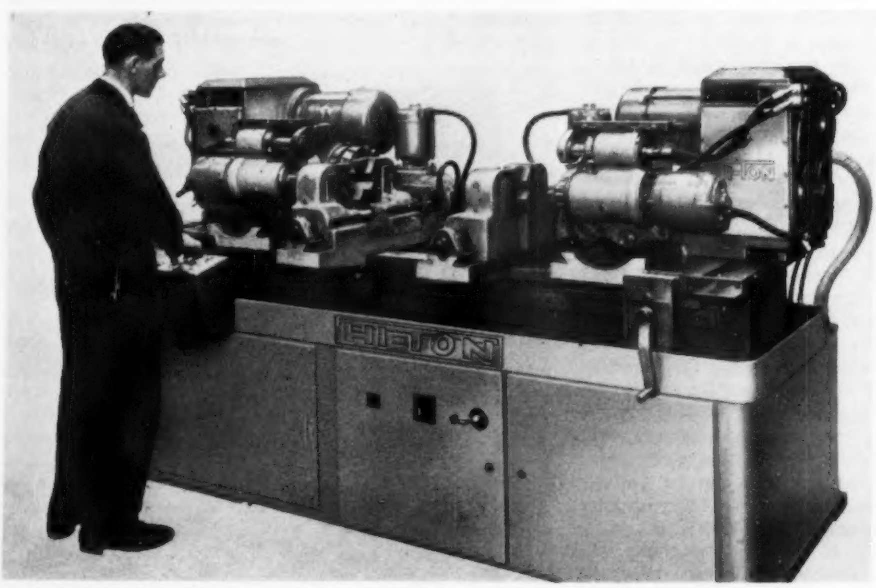
Milling heads are of the mechanical type, each powered by a 5 h.p. motor, but 7·5 h.p. motors can be provided for heavy-duty operation. The left-hand head is fixedly positioned longitudinally on the machine bed, while the right-hand unit is slidable on the bed by means of a rack and pinion motion to adjust its position to suit the length of the workpiece. All spindles run in taper roller bearings, some of which are immersed in oil while the upper ones are lubricated by a pressure feed system. Spindle speeds of from 30 to 110 r.p.m. can be provided by selection of change gears, conveniently located on the top of the head, as shown with the cover removed in the smaller illustration. Hydraulic operation is arranged for the traversing feed movement, which is steplessly variable in rates from 1·0 to 10 in/min. A rapid-approach movement of the cutter is at the rate of 40 in/min and at the end of the machining cycle the heads are rapidly retracted at 50 in/min.

Drilling heads are mounted on the front side of the milling heads. Powered by a direct-coupled 0·8 h.p. motor, the drilling spindle runs in taper roller bearings at a constant speed of 960 r.p.m. and is oil-bath lubricated. An independent hydraulic feed gives fast approach, pre-selected feed rate, and rapid retraction of the spindle.



A milling and drilling head retracted to its standing position. Change gears are accessible on removal of the upper cover plate, as shown

The workpiece is supported by either one or two automatically operated vices, as required. These vices, which are positioned manually between the two milling heads, are fitted with vee-type jaws to suit the component to be handled, and are automatically self-centring. A motorized worm drive actuates the jaws of a vice, which close to a pre-set pressure determined by adjustment of the hydraulic operating



Hi-Ton automatic cycling, facing and centring machine, set up with two work vices

pressure on the worm retainer. The method, in turn, ensures the automatic cut-out of the motor when the required closing torque has been attained. To open the vice, the motor is run in reverse and the limited amount of axial movement required to return the worm to its operating position gives a slight delay which is sufficient to enable the motor to reach full speed and, consequently, to apply adequate torque to release the vice screw.

Housed within the machine base, the hydraulic system is a self-contained unit independently powered by a 3 h.p. motor. Solenoid-operated valves controlling the machine cycle are grouped and accessibly mounted on rigid members of the base frame. Valves controlling the feed rates of milling and drilling heads are similarly grouped on the rear ledge of the base frame. The coolant sump, occupying a central position in the machine base is a separate fabrication. From it, a motorized pump feeds coolant to the milling cutters and centre drills. Flow jets to the milling cutters are integral with the cutter guards. Mounted over the sump is a swarf-collection and drainage tray, and access is given at the rear of the machine for the rapid removal of swarf.

Easy control of the automatic cycle is ensured by the provision of an electrical panel on the left front of the machine. The operating procedure is initiated by first depressing two buttons to secure the workpiece in position and then depressing the cycle start button. This starts rotation of the milling cutters and simultaneously the fast approach of the cutters to the work. Then follows the traverse of the cutters across the end faces of the component at the selected feed rate and, on completion of this operation, a rapid return of the heads to their standing position, which also is the operative position for the drilling heads. The drilling spindles commence rotation, make a fast approach to the work, and perform the centre-drilling operation at the pre-set rate of feed. Automatic retraction of the drilling spindles completes the machine cycle, and depression of a further button releases the vices ready for unloading.

The manufacturers are Hi-Ton Machine Tools Ltd, Birmingham, and sole world distributors are Drummond-Asquith Ltd, King Edward House, New Street, Birmingham.

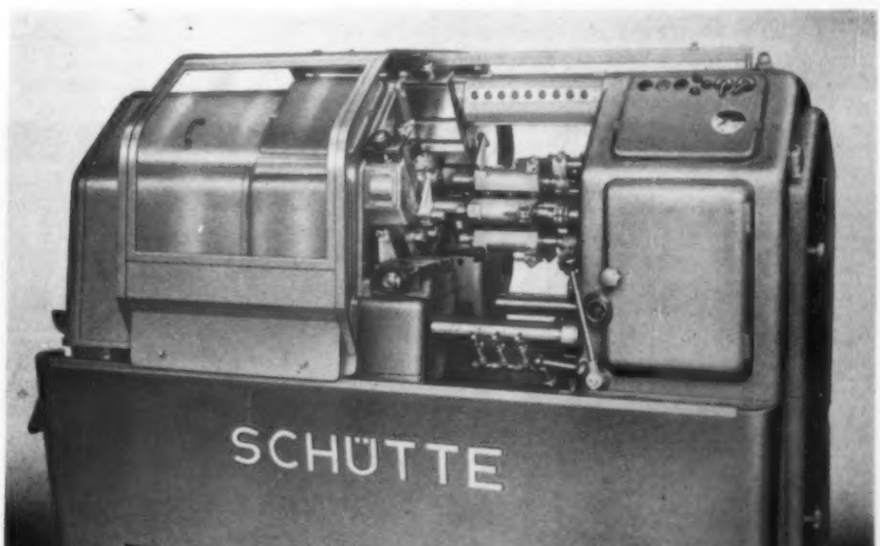
SCHÜTTE TYPE SE 16 AUTOMATIC

A High-speed, Fast-cycling, Six-spindle Machine With a Maximum Bar Capacity of 18 mm Diameter

TO the wide range of multi-spindle automatics built by the firm of Alfred H. Schütte, of Köln-Deutz, Germany, has recently been added a small, six-spindle machine, designated Type SE 16. Embodying the well-proved basic design principles of the Company's SD range of automatics, the new machine incorporates other interesting features. Due to the higher spindle speeds employed, the shorter indexing times, and the reduced overall dimensions, however, it may be regarded as a completely new design. An outstanding feature is that the longitudinal tool carriers are designed as quills instead of the usual slides. It can handle round bar up to 18 mm ($\frac{23}{32}$ in) diameter, hexagon bar 16 mm ($\frac{5}{8}$ in) across flats, and square bar 13 mm ($\frac{1}{2}$ in) across flats. Maximum bar feed length is $4\frac{3}{4}$ in.

Notwithstanding its compact design—the machine is 8 ft 10 in long × 3 ft 7 in wide—the frame is heavily and rigidly constructed to allow the use of high speeds and rapid motions without vibration, and weighs approximately 4 tons. Bar guides add 7 ft 10 in, plus 3 ft required for retraction, to the overall length, and 700 lb to the weight. Both the machine base and the spindle carrier housing are carefully designed to facilitate the flow of swarf to a large drip pan lying along the longitudinal axis of the machine. Removal of swarf from this pan can be made without interruption of machine operation.

The wide range of 45 spindle speeds from 400 r.p.m. to 5,000 r.p.m. allows optimum cutting speeds to be selected for any material being machined and also for the use of carbide-tipped tools. Piece times of from 2 sec to 41 sec make possible the economical production of even simple components. Should the machine be required to operate as a double three-spindle machine, an additional bar feed



Schütte six-spindle Type SE 16 automatic, featuring quills instead of tool slides, and a cross slide at each station. Bar capacity is 18 mm

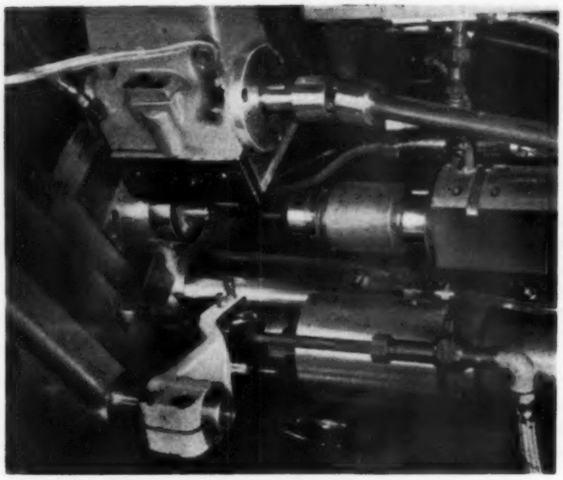
and chucking arrangement, with bar stop, is fitted at the third spindle position.

A 10 h.p. motor housed in the base powers the machine, the drive being by multiple vee-belts to the main shaft from which all other motions are derived. Power is transmitted to the work spindles, running in precision-type anti-friction bearings, through intermediate gears, change gears, a central shaft and drive gears. Camshafts are arranged at the front and rear of the machine and driven through change gears, rapid traverse and feed traverse clutches, and worm shafts. Coupled to the feed clutch is a disc-type brake which stops the worm shaft when the feed is disengaged. The camshafts provide the motions for the cross slides, the quill traverses, spindle drum indexing and interlocking, bar feed and chucking, positioning of the bar stop, and the remaining control movements. Any one of the many special attachments can be fitted and driven at any spindle position by a central gear set built into the machine drive head. Pumps are driven from the intermediate gears of the spindle drive.

All the change gears for the spindle speeds, cycle times, and the built-in drive for screw threading operations are readily accessible on removal of a cover door on the drive gear housing. Similar accessibility is arranged for the quadrant levers for the cross slides and longitudinal quill traverses, and the Geneva motions for indexing the spindle drum and the bar guides. Two vertical locking bolts, operated hydraulically and by spring pressure, interlock the spindle drum. Each spindle has an independent cross slide, the traverse of which can be limited by adjustable stop screws. The spindle collets are of the draw type.

As already noted, individually movable quills are fitted instead of the conventional longitudinal tool slides. Quill holders are clamped to the carrier block in an adjusted axial position. Thus, each spindle has a cross slide with independent feed and a longitudinal tool carrier with independent traverse. Hardened and ground quills and quill sleeves are provided with coarse and fine adjustment, and travel is limited by end stops. Quill feed is by means of push rods, actuated through cams and quadrant levers. In addition to a push rod for each spindle, two extra rods can be fitted for further longitudinal movements. If necessary, in order to safeguard sensitive tooling against overloading, special safety-type push rods can replace the standard solid push rods.

The bar guide assembly comprises a pedestal base, in which change gears, tools, and equipment are housed, and



Work area, showing pick-off device, right, and tools mounted in bar stop

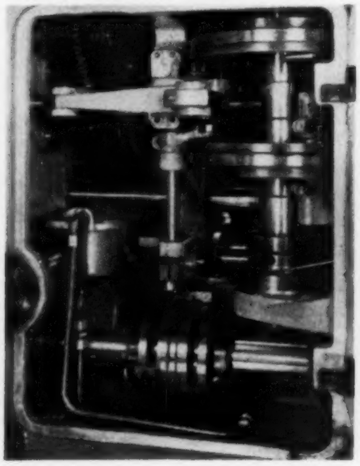
the guide tubes. These tubes are individually retractable to permit easy removal when feed fingers need to be changed. Drive for the bar guide assembly is not from the spindle drum but by an individual Geneva motion that can be disengaged independently of the indexing mechanism of the spindle drum. By this method, new bar stock can be loaded into the guide tubes while the machine is running. Bar feed length is adjustable by means of a quadrant lever.

Lubrication is by a pressure system delivering filtered oil to various points on the machine, and a signal light indicates the system is functioning at correct pressure. For spindle speeds in excess of 3,000 r.p.m. a supplementary pump and delivery system is brought into operation to provide the additional lubrication necessary for the work spindles. Other and less critical lubrication points are fed with oil by gravity from high-level reservoirs.

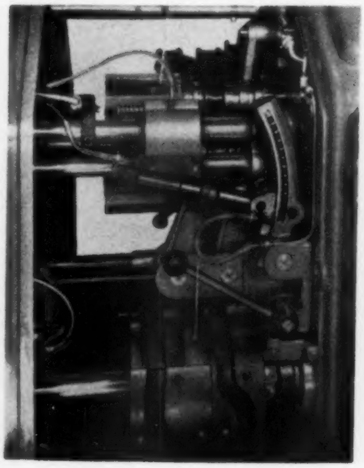
Operational reliability is enhanced by the provision of numerous safety devices. The machine is automatically stopped should:

the feed drive be overloaded
the bar stop be prevented from swinging into position
the work spindle drum fail to lock in its indexed position
a spindle collet be not correctly closed

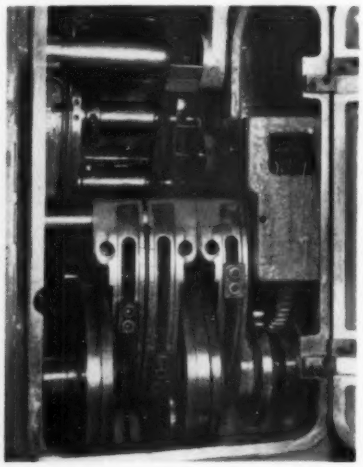
Cams on vertical shaft for quill movement

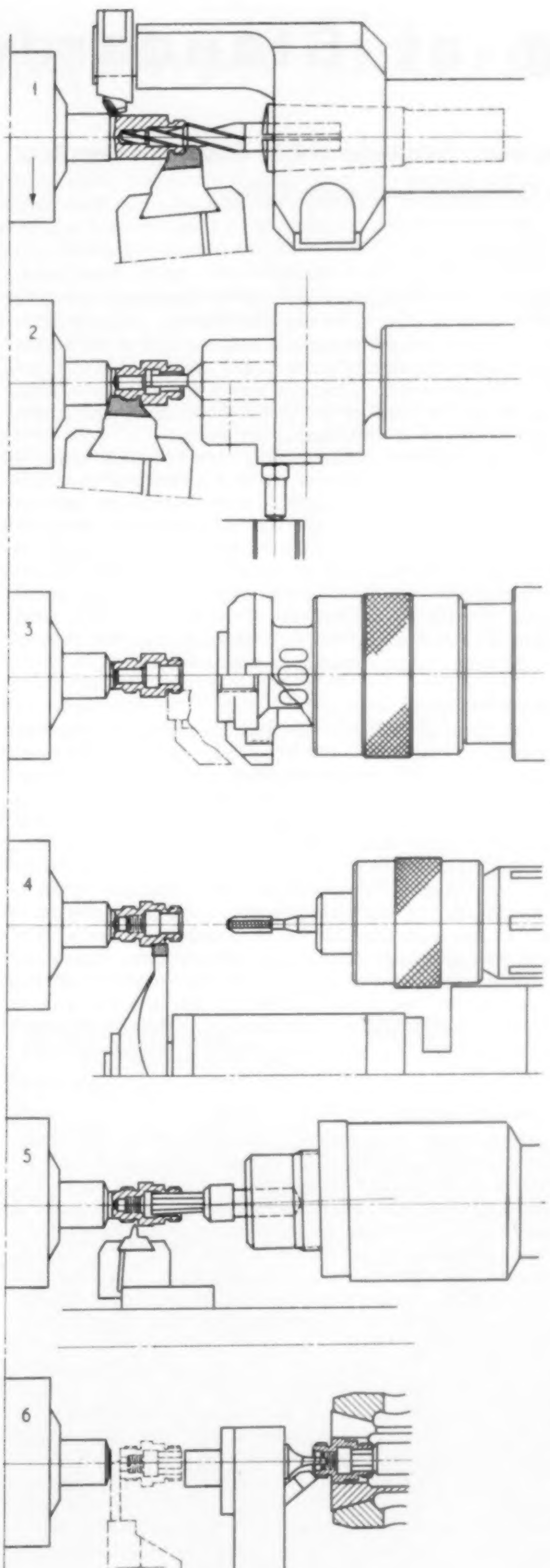


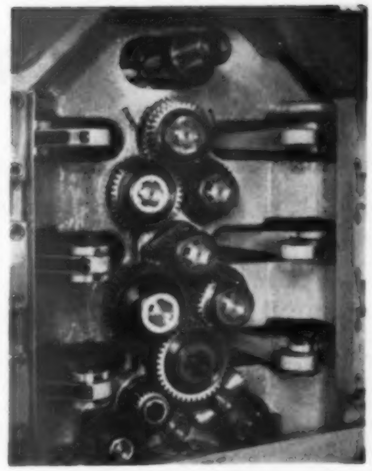
Quadrant for adjustment of bar feed length



Cams and adjusting levers for cross slides







Change gears for spindle speeds, cycle timing, and threading drives

the bar stock in one of the feed tubes be exhausted
a safety-type quill push rod be overloaded
the pressure lubrication fail to function correctly
the oil reservoir require replenishing.

A row of signal lamps arranged above the working area indicates the location of any source of trouble and a red light draws attention to the stoppage of the machine.

To facilitate setting, a dial indicator is provided adjacent to the duplicated controls on the front and the rear of the drive head. One revolution of the pointer corresponds with one revolution of the camshaft. The dial, graduated in degrees, is divided into two sectors of rapid traverse and work traverse. Rapid traverse is subdivided into three sectors of return, indexing, and approach. In the return sector is marked a small sector defining the included angle during which the bar collets are open. Thus the pointer continuously indicates the operational position in the cycle.

Many auxiliary attachments are available and extend the field of application of the machine. They include attachments for threading, high-speed drilling of small diameter holes, pick-off for machining the part-off end of the workpiece, longitudinal turning behind shoulders, eccentric turning, rotating milling, cross-hole drilling, marking, internal polygon shearing, and external polygon turning. Every attachment can be used in any spindle position. Since they are all automatically lubricated, most of them can be used at the highest practical spindle speeds.

The tooling drawing shows the set-up to produce a screwed brass component, approximately $\frac{3}{8}$ in diameter and $\frac{7}{8}$ in long, for the purpose of demonstration. With the machine spindles running at 4,000 r.p.m, the piece time is 3 seconds. Operations at the six stations are as follows:

(1) Turn O.D. with carbide-tipped tool; drill with 3-diameter stepped twist drill; turn front end and undercut with carbide-tipped form tool

(2) Turn rear end and undercut with carbide-tipped form tool; bore recess in drilled hole

(3) Screw front end, left-hand, with button die

(4) Form hexagon with carbide-tipped, polygon-turning tool; tap drilled hole

(5) Screw rear end, right-hand, with thread-chasing attachment; form internal hexagon with internal polygon-shearing attachment

(6) Part and pick-off; counterbore rear end with flat drill and chamfer with tool bit, both mounted on rear of the bar stop plate.

Sole selling agents in the United Kingdom for Alfred H. Schütte machine tools of all types are Rockwell Machine Tool Co. Ltd, Welsh Harp, Edgware Road, London, N.W.2.

Data Processing at Standard-

Recently Installed at Canley is a Leo Digital Computer, Supplemented by Friden Flexowriters and Other Processing Equipment, to Constitute a Complete Automatic Office

WHEN after preliminary survey and consideration a decision is made to acquire a computer for data processing and accounting in a plant manufacturing a range of complex products, the building and installation of the computer and the introduction of the system cannot be lightly or rapidly undertaken. Although embodying standardized component units, a computer is designed specially to meet specific requirements and commonly a period of about two years is taken to build and install. Obviously, the first and most important task is accurately to define and co-ordinate all the divergent, and sometimes conflicting, requirements of a number of manufacturing plants, suppliers, distributors, stores and departments. Only when this has been virtually completed can the design be finalized and construction commenced.

Experience at the Standard-Triumph plant provides a typical example. In May 1957, faced with the ageing of the existing punched-card machinery, the consequent impending need for replacement, and a rising volume of production, the management decided to institute a survey of requirements and investigate the potentialities of a computer system. A "Computer Investigating Committee" was set up, consisting of management representatives of several departments.

In May 1958, the decision was taken by the management to purchase a Leo II C computer as best suited to its needs. Actually, it was on a larger scale than was strictly necessary for the then current needs, but the Company's expansion programme was under way and the computer selected had to be capable of handling an increased volume of work in the future. The first task was to ascertain the requirements of the various departments, and to reconcile these by adjustment and co-ordination into an integrated whole, in order to facilitate programming and avoid duplication and overlapping of results.

This work was undertaken by a "Steering Committee", made up of a senior member of each department under the chairmanship of a Board director, which sorted the requirements of all departments, adjusted contacts and the transfer of information to the various manufacturing plants, found solutions to problems arising and formulated an overall pattern. The computer department — the staff for which had undergone a full course of training with the computer manufacturers—expanded and detailed this pattern to the form from which computer programmes could be written. About eight months was occupied in completing this task, since some departments required information on a day-to-day basis and others week-by-week or even month-by-month.

While the computer department proceeded with planning and programming, the methods and systems department was engaged in organizing the system of communications between departments and between the different plants. It had to decide the most suitable methods to employ for specific duties — whether punched cards, punched tapes, or telefacsimile—and the equipment required, and also to plan, section by section, for the smooth changeover of procedure.

The digital computer functions by being fed with information in simple terms and performing simple tasks with extreme rapidity. Suitably programmed in simple, step-by-step terms, a complicated task can be completed in a very short space of time. Leo can read standard punched cards at the rate of 400 per minute, or punched tape at 400 figures per second, and internally the "steps" can be made at a rate of 2,000 per second. Magnetic tape, for reading or writing, can be run at 7 inches per second (35 ft/min), and at this speed it can carry 4,000 figures per second.

In an automobile manufacturing plant the major tasks for the computer are stock control and scheduling the build programme. Each is staggering in its magnitude: the one from the number and diversity of the parts, both internally produced and purchased from outside suppliers, needed to build an automobile; and the other from the huge number of variations to the basic specification of a particular model made possible by the exercise of options in colour, trim, equipment and accessories, and multiplied by the number of different models currently in production.

Computer input

The computer handles the large quantities and different qualities of data by a system of multiple input channels. Each channel has an independently operating reading unit feeding the computer through a buffer store. The reading unit reads ahead of the computer, which draws the data from the buffer store as required. Three such channels, reading simultaneously, form an effective combination suitable for large scale operations. For example, brought-forward, current, and amending data can be read in parallel and all data made available for calculation immediately. The buffer store of each channel can hold the entire contents of an 80-column punched card. Paper tape readers and card readers share two input channels, so that any two may be linked to the computer simultaneously. Magnetic tape units have one input channel which handles up to four magnetic tape decks.

Computer output

Output from the computer is required in different forms: for immediate use, for carry-forward information, and for results which require further processing before use. The problems presented are, therefore, comparable to those of input, and three input channels are employed. The channels feed, respectively, a Hollerith card punch, a Powers Samastronic printer, and up to four Decca magnetic tape decks. Each channel has its own buffer store with a capacity of a complete block of 16 compartments on magnetic tape, an entire card, or up to 99 mixed characters for printing.

Results are output on the channel best suited to the particular purpose. Carry-forward results are normally basic records brought up to date during the computer run. If so, they are recorded on magnetic tape since these records are cumbersome by comparison with current data and operational results. Direct printing is used when finished results are produced for information and action and when these results are not required for further processing. Where the results require sorting into a different sequence before being printed (normally in small volumes), or are to be used for further data processing functions, cards are punched.

Triumph International

Verification of cards is by means of automatically punched parity digits, which are checked back at a sensing station immediately the punching is completed. Magnetic tape output is fully checked by comparing the newly written tape with information held in the buffer store. In the event of a tape check failing, the computer is not delayed. The information is automatically rewritten and the faulty block is marked. No computer time is lost in carrying out the automatic checks.

Storage

The main store comprises matrices of magnetic cores, access to which is controlled by transistor circuits. All transfers from this store are automatically checked by parity digits to ensure instantly that the instruction or the number concerned has been retained correctly. The time taken to gain access to the store is fifteen-millionths of a second. For clerical work, a large, fast store is essential since even in a short run reference to the store may be made several millions of times. Clerical programmes demand random reference by reason of many logical variations that have to be taken into account and continual reference made to general information.

The magnetic core store consists of 8,192 compartments. This large store capacity is usually adequate for internal storage of all kinds—programme, information tables, and intermediate results. Should greater capacity be required, further large magnetic core stores can be added.

Arithmetic unit

Another factor affecting the speed at which the computer can work is the speed of the arithmetical operations. Addition, subtraction, multiplication, and division are carried out in special circuits in an arithmetic unit. Addition or subtraction of up to twelve-digit numbers is effected in four ten-thousandths of a second. The time required for multiplication varies according to the size of the numbers involved but in most instances operations can be performed at a rate of 1,000 per second. Automatic division provides both the quotient and the remainder at a rate of 300 per second.

Fourteen special immediate-access registers in the arithmetic unit can each hold up to a twelve digit number. One pair forms the main accumulator and can hold the double-length result of a multiplication. Another forms a subsidiary accumulator. There are three separate modifying registers, each of which can be stepped on by positive or negative increment. Repetitive processes are thus made completely automatic.

Instruction code

Speed of working is also influenced by the range of the instruction code. Invaluable experience, accumulated in practical operation over the past eight years, is reflected in the simple but comprehensive code developed for the Leo computer. It offers the full range of arithmetic facilities and includes, also, many instructions designed for data processing operations. Each instruction is framed to ensure the performance of specific operations in the most direct and logical manner, in order to save programming effort, computing time, and storage space.

Special instructions, some of which have already been briefly noted, include input, output, conversion, division and modification. A single instruction effects conversion from binary to sterling or decimal notation. The divide instruction automatically takes care of the sign and produces a quotient and remainder.

One simple instruction of great utility, designated "Augment", combines the two processes of addition and

A Leo computer installation similar to that installed at Canley. Central is the computer control panel, behind which are the several rows of computer racks. On the extreme left are magnetic-tape units and, in front of them, the card feeds. To the right are the printers and the card punches

The marshalling area on the top floor of the new assembly hall accommodates up to 500 finished bodies

transfer and was designed to simplify and speed the work entailed in producing totals. It has proved to be remarkably effective in so-called "clerical" work where many totals have to be formed at each stage of the operation. Another instruction produces totals of a different kind. By adding together numbers, in blocks of variable length, it forms a negative sum that is of great convenience in data-processing operations.

Inspect, read, and write instructions complement the system of multiple input and output channels. They enable blocks of variable length to be input and output, to and from the rapid access store. By this means, different streams of information can be handled individually, and programmes to be arranged with the minimum amount of information transfer work. The computer can translate a programme from the simple terms in which it is prepared to the more complex terms by which it is obeyed. This automatic translation has to be done only once; subsequently it is available for immediate use as and when required. Thus, programming is greatly simplified and the full potential of the computer can be realized.

Auxiliary processing equipment

A computer requires auxiliary apparatus to:
(1) convert written programmes into coded information on a punched tape or a magnetic tape and to check the accuracy of the conversion
(2) feed the information automatically into the computer at high speed
(3) record the computer output in coded form
(4) convert the coded output into printed characters.
Commercially available punched-tape and teleprinter techniques will satisfy the primarily important input and output requirements and can also provide a variety of auxiliary information-handling facilities.

At Standard-Triumph the Friden Flexowriter, a high-speed automatic writing machine which operates from and also produces punched paper tape, is the keyboard machine by which the integrated, data-processing system is established. Additional to the familiar advantages of an electric typewriter, the Flexowriter incorporates programmatic features and a built-in tape reader and a tape punch. The reader functions to sense codes—combinations of holes punched in a paper tape—and translate them into clear type or a further punching of tape, at a speed of 572 codes per minute. The punch, when switched on, will automatically encode and punch into tape or into edge-punched cards, whatever is either manually typed from the keyboard or automatically read by the reader unit. The production of this tape does not retard the speed of the typing operation in any way.

Closed-circuit TV monitor screens in the basement control room enable body progress to be checked

Operator's errors can readily be corrected in the tape by merely turning back the appropriate codes in the punch unit and depressing the "Code Delete" switch. All these codes are then obliterated by perforating the tape or card in all possible channels, and the deleted codes will automatically be ignored in subsequent passages through the reader unit. The writing machine can itself be programmed and automatically controlled by a master tape or edge-punched card, thus reducing manual operations to the minimum.

The input keyboard is used solely for entering new or variable data not previously recorded. All other repetitive data are automatically typed by the depression of the "Start Read" switch. Data is read from the tape or card on which it is stored as a permanent record. Auxiliary units, such as a tape or edge-card reader, can be cable connected to a programmatic model Flexowriter, enabling the machine to read selected data from two originating tapes or cards under the control of codes in the programme. An auxiliary punch makes it possible for the machine to produce two different tapes as an automatic by-product of preparing the original document. The master card selects, automatically, data to be punched into either or both by-product tapes.

Friden Programmatic Flexowriter, a high-speed, automatic, writing machine

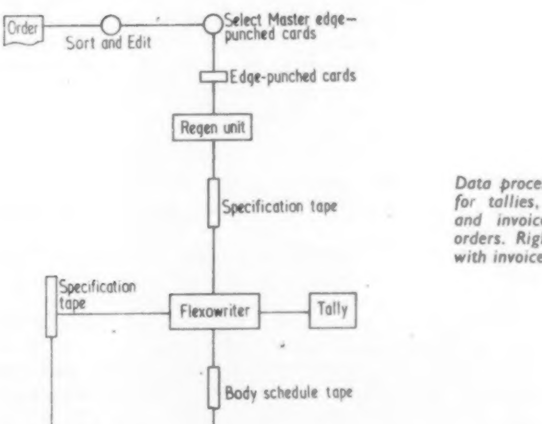
Initial requirements

As originally postulated, the equipment was required to handle, process, document, and record data as follows:

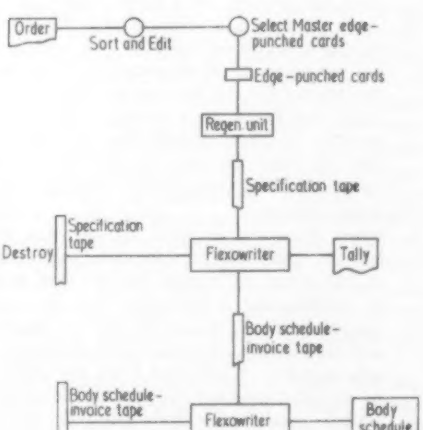
(1) Volume
- More than 600 production "tallies" per day

(2) Record on documents
- (a) Tallies,
  - Model
  - Model code
  - Production group
  - Destination and order number
  - Destination code
  - Exterior colour and code
  - Trim colour and code
  - Tyre size and code
  - Tyre type and code
  - Extras and codes
  - Special instructions
- (b) Body schedules,
  - Rota number
  - Production group
  - Destination and order number
  - Exterior colour
  - Trim colour
  - Tyre size
  - Tyre type
  - Extras
  - Special instructions
- (c) Invoices (home),
  - Data as for tallies, with the exception of "Model code" and "Production group".
- (d) Invoices (export),
  - Data as for "Body schedules", with additional variable and pricing data.

(3) Leo computer, by-product tape
- Rota number
- Model code



Data processing flow charts for tallies, body schedules, and invoices. Left: Home orders. Right: Export orders, with invoice tape for mailing



Production group
Destination code
Exterior colour code
Trim colour code
Tyre size code
Tyre type code
Extras codes

(4) Master edge-punched cards
Standard data to be recorded on each master card:
(a) Model cards,
Model description
Model code
Production group
Destination code (home only)
Exterior colour and code
Trim colour and code
Tyre size and code
"Stop codes" to be programmed into the model cards to allow the operator to key in manually the following data at predetermined positions on the tally:
Production group (if variable)
Destination and order number
Destination code (export only)
(b) Tyre type cards,
Tyre type and code
(c) Extras cards,
Extra and code

Procedure

The operational procedure was devised with the object of providing for the complete integration of Tally, Body schedule, and Invoice routines. An important aspect of the scheme is the elimination of the repetitive typing of like data on tallies, schedules and invoices, thus minimizing the hazard of transcription errors. As operational prerequisites, a file of Master edge-punched cards covering all practical combinations of Models, Tyre size, Tyre type, and Extras was set up and, where necessary, existing documentation was modified to achieve compatibility throughout the system.

Sequentially, the operational procedure is as follows:

(1) Order handling
A Friden Regenerator unit is used to collate individual edge-punched cards or punched tapes and reproduce their information on a single length of punched paper tape. Incoming orders are passed through the existing sorting and editing routines. On completion, the orders are passed to the Regenerator operator who selects from the Master file the appropriate Model, Tyre, and Extras cards and feeds them through the regeneration unit to produce one Specification tape for each vehicle ordered. The specification tape is attached to the order and passed to the Flexowriter operator for tally writing.

(2) Tallies (home)
From the order received, the operator feeds the Specification tape into the Reader, recording automatically at 100 words per minute the standard data, and stopping at predetermined positions for the manual entry of variable data. The output tape or card listing the Body schedule details, obtained as an automatic by-product of tally writing, is removed from the punch and attached to the tally, together with the Specification tape. Should tape be used for the Body schedule, the data is recorded on a tape of different colour from that of the Specification tape for ready identification.

(3) Tallies (export)
The same basic procedure as for Home orders is applied to Export orders, with the exception that once the tally is produced the Specification tape is destroyed.

(4) Tallies (bulk orders)
For orders calling for bulk quantities of one specific vehicle, the Regenerator operator produces one Specification edge-punched card recording the quantity. By reference to this number the Flexowriter operator feeds the edge-punched card through the Reader the appropriate number of times to produce one tally and one Body schedule tape per vehicle.

During the period between the production of tallies and the Body schedule, each tally is allocated a Rota number which is recorded both on the tally and the Body schedule tape. As becomes necessary as the tallies accumulate, they are marshalled into the required sequence for production of the Body schedule. The Flexowriter operator detaches the Body schedule tape from the tally and feeds it into the Reader, automatically recording standard data and manually adding the Rota number for each vehicle. The Body schedule is retained for possible application in producing the Build Programme (home) and Build Programme and Invoice (export). As an automatic by-product of the operation, an output tape is produced containing selected data (as noted earlier) for automatic conversion into punched cards, and providing an input to the Leo computer.

Since the format of Invoices (home) is almost identical with that of the tally, use is made of the Specification tape to produce these documents. It becomes possible to produce the invoice (excluding the pricing data) after the Rota number has been allocated. The Invoice operator detaches the Specification tape from the tally and feeds it into the Reader to record standard data; variable data is added manually by keyboard operation. Subsequently, the Specification tape is destroyed and the Invoice set retained to await the release of the vehicle, when it is completed by addition of the pricing data.

To enable an Invoice (export) to carry several vehicles, the format of this document is identical to that of the Body schedule, except for relatively insignificant variations. The Body schedule tape, consequently, is best suited to produce the individual vehicle data on the Invoice master. It is not possible to produce Invoices (export) until the vehicles are ready for release. A Release invoice is raised by manually typing the variable heading information, and then feeding in the Body schedule tape for each vehicle. The Release invoice is produced with the punch control switch at "All" to provide a composite tape, recording all the data on the Release invoice, which is held pending pricing. For completion of the Invoice, the composite tape is run through the Reader under "Non-Print" condition and, with the punch control at "All", the pricing data is added to the Invoice master. In addition to the completed Invoice master, this operation will provide a final composite Invoice tape which will be available for mailing overseas.

Advantages of the system

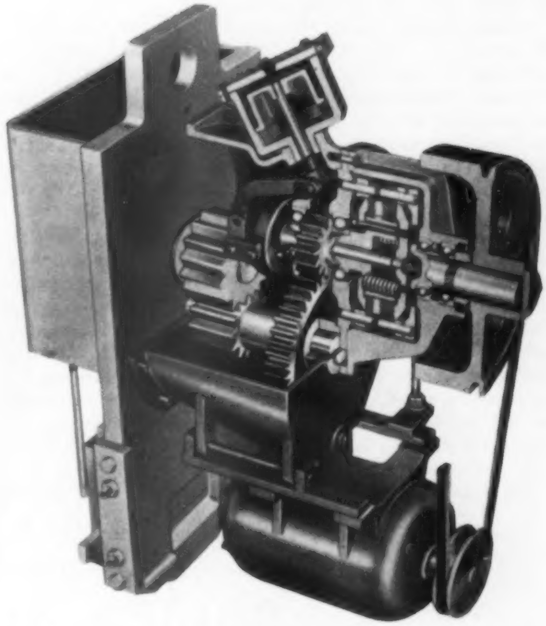
For the processing system outlined, the following advantages are claimed:

1. Highly efficient and mechanized documentation of tallies, body schedules, and invoices
2. Increased accuracy, as no transcription errors
3. Elimination of the need for the repetitive typing of like data or tallies, body schedules, and invoices
4. Complete integration of sales documents with the additional advantage of an error-free, automatic by-product tape for computer intake.

The system is, of course, not rigid. It is, in fact, exceptionally versatile and can readily be modified to accommodate additional or reduced requirements or to meet changed conditions. Currently, processing is carried out on 8-channel tape equipment. Should future requirements necessitate a greater volume of direct tape input to the computer the more flexible 8-channel tape can be readily converted to 5-channel tape by a standard Tape Converter.

British Clearing Torc-Pac Presses

A Range of New Open-backed, Inclinable Presses Manufactured by Vickers-Armstrongs Ltd, at Crayford



Sectional view of the sealed, interchangeable Torc-Pac drive unit

SINCE 1946 Vickers-Armstrongs (Engineers) Ltd. have been building large mechanical and hydraulic presses at their Newcastle-on-Tyne works to the designs of the Clearing Machine Corporation, a division of United States Industries Inc. of Chicago. Additionally, Vickers-Armstrongs are now manufacturing a range of three smaller presses, in 22, 32, and 45 ton sizes, at their Crayford works. Of the open-backed, inclinable type, all are of modern design with frames of all-steel welded construction providing great resistance to deflection under load. A feature of each press is the redesigned, pneumatically operated "Torc-Pac" drive unit incorporating an air-operated friction clutch and brake that, since it requires no adjustment, is sealed. The complete drive unit, including the flywheel, is located between the frame side members, resulting in a compact design conducive to safety and occupying a minimum of floor space. In the case of the smallest unit, the 22 ton model, the necesary floor space is 28 in × 39 in when the press is in the vertical position.

Clutch, brake, flywheel bearings and gearing run in an oil bath. Friction plates are of sintered bronze and subject to a lower rate of wear than the plates of conventional clutches as engagement is effected through entrapped oil which is placed in shear. Furthermore, the sintered plates are much less likely to be affected by high-temperature operating conditions. A mechanical interlock prevents simultaneous engagement of the clutch and the brake; air pressure releases the brake and engages the clutch. When air pressure is exhausted from the clutch control, spring pressure releases the clutch and actuates the brake. In the event of either a compressed air or an electrical failure, the brake is applied and the press brought to a safe stop.

A 45 ton British Clearing Torc-Pac press, the largest of the three models

From the clutch the drive is taken through a spur-type reduction gear to a precision-ground eccentric shaft supported, immediately adjacent to the eccentric and largely eliminating the bending moments occurring in the conventional crankshaft design, in anti-friction bearings. The shaft is arranged at 90 deg to the usual layout and projects through the front housing to provide a direct power take-off for slide feeds or other auxiliary equipment. A stroke-indicating arrow shows the position of the eccentric and fitted above the shaft extension is a limit switch actuated by an adjustable cam to stop the press with the slide in the uppermost position at various operating speeds.

The pitman has a bifurcated lower end to which is fitted a stout cross-piece. Passing through this member is the pitman extension, secured in position by deep lock nuts above and below. The upper nut is graduated in divisions of 0·001 in for the rapid and accurate setting of the press slide; the range of adjustment varies from 2 in to 2½ in depending on the size of the press. This design of pitman results in the press load being transmitted by full-length contact of the deep nut, whatever the adjusted position. The slide is guided in vee ways, with one fixed and one adjustable gib strip. To obtain relatively quiet operation,

the knockout bar is spring loaded in the top position. Lubrication of the running gear—eccentric-shaft bearings, pitman bearings, and slides—is from a single point on the press frame and feed can be arranged by oil gun, manually operated pump, or automatic pump.

Bed and bolster sizes for the three models are 21 in × 11 in, 24 in × 15 in, and 28 in × 18 in respectively, with bed openings of 9 in × 5 in, 11 in × 8 in, and 14 in × 11 in. As standard, plain bolster plates are supplied but plates can be machined to meet specific requirements. The presses are readily inclinable by means of an adjusting screw operated by a wrench. Possible angle of inclination is up to 25 deg on the 45 ton press and up to 30 deg on the two smaller models.

A dust-tight and oil-tight control cabinet containing the motor starter, relays, transformer, and fuses is mounted on the side of the press frame. The standard selector switch on the front of this cabinet has four positions: Continuous, Off, Inch, and Once. Alternatively, the American Standard selector switch having an additional position "Machine continues to top of stroke" can be fitted. As standard, the operator's controls comprise Run and Stop buttons mounted at the right-hand front of the bed. Foot switches and additional controls are available and can be simply plugged into the control circuit, which operates on 110 V.

Of the three models in production, the two smaller machines, powered by 2 h.p. and 3 h.p. continuously rated motors respectively, are each available with a choice of various fixed speeds or with a variable-speed drive. Maximum operating speeds of these models is 225 and 150 stroke/min. The 45 ton press is supplied only with fixed speeds of 75, 90, or 122 stroke/min, the drive being by a 5 h.p. continuously rated motor. All models are built with any one of four stroke lengths, with a maximum stroke of 4 in, 5 in, or 6 in depending upon the press capacity.

Standard and optionally high bodies are built for all three sizes, and the 32 ton and 45 ton presses are available with standard (8 in or 9½ in) or specially deep throat depths (10½ in or 12 in). Other variations can be made to meet special requirements since the frame is fabricated from steel plate. A full range of auxiliary equipment is available for all models, including pneumatic die cushions, slide feeds, stock straighteners, and reels.

For the 32 ton and 45 ton presses, flanged slides with or without slide caps, and adjustable air counterbalance cylinders can be supplied. The sealed Torc-Pac drive unit is interchangeable and need never be opened up for servicing. A stock of rebuilt and guaranteed units is held to maintain an immediate replacement on an exchange scheme.

Rockwell Machine Tool Co. Ltd, of Welsh Harp, Edgware Road, London, N.W.2, are the sole selling agents in the United Kingdom for Clearing Machine Corp. presses.

Copy-milling Combustion Cavities

TO enable rapid two-dimensional machining of combustion chamber cavities in a cylinder head to be carried out, Hayes Engineers (Leeds) Ltd, of Gelderd Road, Leeds 12, have developed a milling machine that is a modified version of their established Tracemaster TM 434-3D model. In this new version, the standard milling head is replaced by a three-spindle head, and the spindles are spaced so as to machine alternate cavities in the head of a 6-cylinder engine. By this means, the finishing of the six cavities calls for only two machining operations, and between these operations the milling head has to traverse only the distance that separates two adjacent cavities.

The template that determines the profile of each cavity is mounted at one end of the worktable, immediately underneath the stylus of a "Digitrace" tracing valve, and this profile is reproduced by the hydraulically controlled movements of the milling head as the operator guides the stylus around the template. At the beginning of the cycle, a cylinder head is mounted in its fixture on the worktable; after the starting button has been pressed, the knee assembly rises to working level. There is a shield plate above the template, and the end of the stylus is tapered: these two features ensure that the end of the stylus is guided into the template orifice. Additionally, the height to which the table rises is determined by the stylus contacting the template, so that the cutters sink to their correct depth in the cavities.

After the first group of three cavities has been milled, the table is lowered just sufficiently to allow the cutters to clear the top of the workpiece as they are traversed to the next three cavities. The stylus is then clear of the template but not of the shield, so the operator guides the stylus through a channel in the shield plate to the second orifice in the template. Following the completion of the second milling operation, the stop button is pressed, the knee falls away, the spindles stop, and the taper end of the stylus remains loosely registered in the guide plate, ready for the machining of the next cylinder head after unloading and reloading.

Conveyorized Furnace

A FURNACE for the heating of light alloy billets prior to their undergoing an extrusion or forging operation has been recently introduced by AEI-Birlec Ltd, Tyburn Road, Birmingham, 24. It is designated the Birlec-Granco furnace, and has become available in Great Britain, Europe and the Commonwealth countries by virtue of an agreement between this firm and Granco Inc, of U.S.A.

The furnace is arranged horizontally, and is a gas-fired rapid heating unit incorporating a chain conveyor that moves intermittently, discharging individual billets at a rate that can be adjusted to suit requirements. Billets are fed into the furnace automatically from magazines and are carried, on attachments projecting upwards from the conveyor, through a slot at the bottom of the heating chamber. Gas burners along both sides of the chamber heat the billets rapidly but uniformly, and temperatures are controlled by Honeywell Brown electronic regulators connected to thermocouples giving direct readings from the billets. Thus, the metal cannot be overheated and can be held at a constant temperature during interruptions in production.

Mechanized handling of the billets, at the delivery end of the furnace, is controlled by push-button, and one operator, using a single control panel, can supervise both this and the magazine feed. The heating chamber is lined with reinforced refractory material, and is built of a number of sections so that its overall length can be made to suit any requirement. Heating chambers are available in standard cross-sectional sizes for billets of the following diameters:— 5 to 7 in; 6 to 9 in; 9 to 14 in and 11 to 16 in; chambers of larger or smaller sizes can be supplied, if required.

Heating times vary according to billet size, type of alloy and temperature required, but from a cold-furnace start an appropriately heated billet can be delivered in 10-30 min. The standard burner fittings are designed for the combustion of town gas, but others can be supplied if the customer's preference is for natural gas or various other types of fuel.

In a manner of speaking, to fit Hoffmann bearings is a matter of choice inasmuch as Hoffmann journal bearings are made with four ranges of diametral clearance to suit differing working conditions. These are known as 'dot fits' and identified by the number of polished circles on the stamped side of the outer race.

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'OOO'—For use when both races of a bearing are made an interference fit, or when only one race is an interference fit and there is loss of clearance due to temperature differences.

'OOOO'—These bearings are used where there will be some loss of clearance due to temperature differences and both races must be an interference fit. It is recommended however, that customers should always consult us before ordering bearings of this fit.

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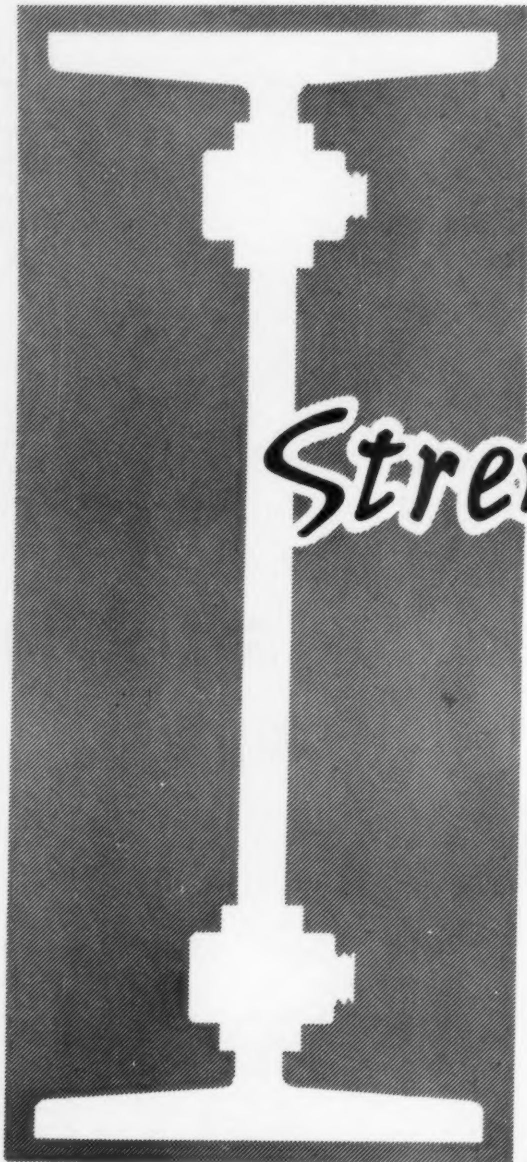

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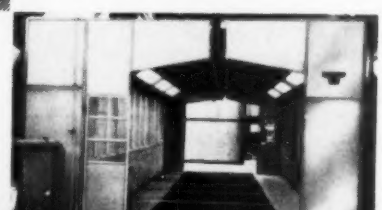

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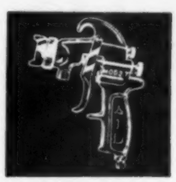

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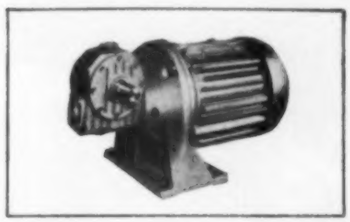

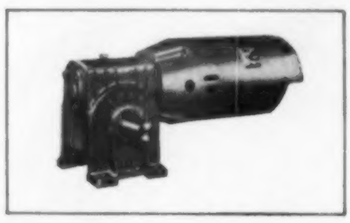

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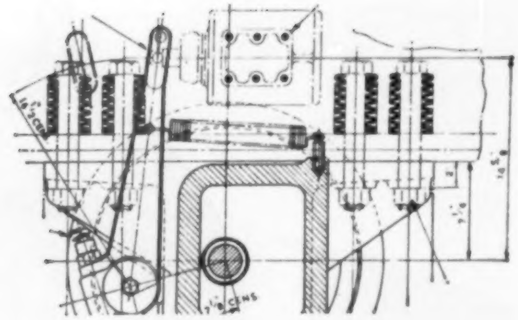

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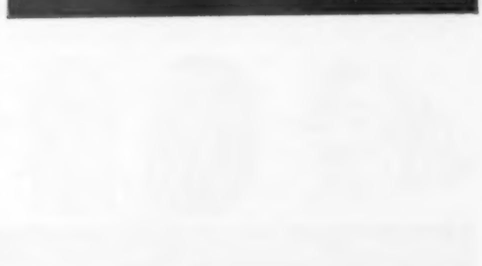

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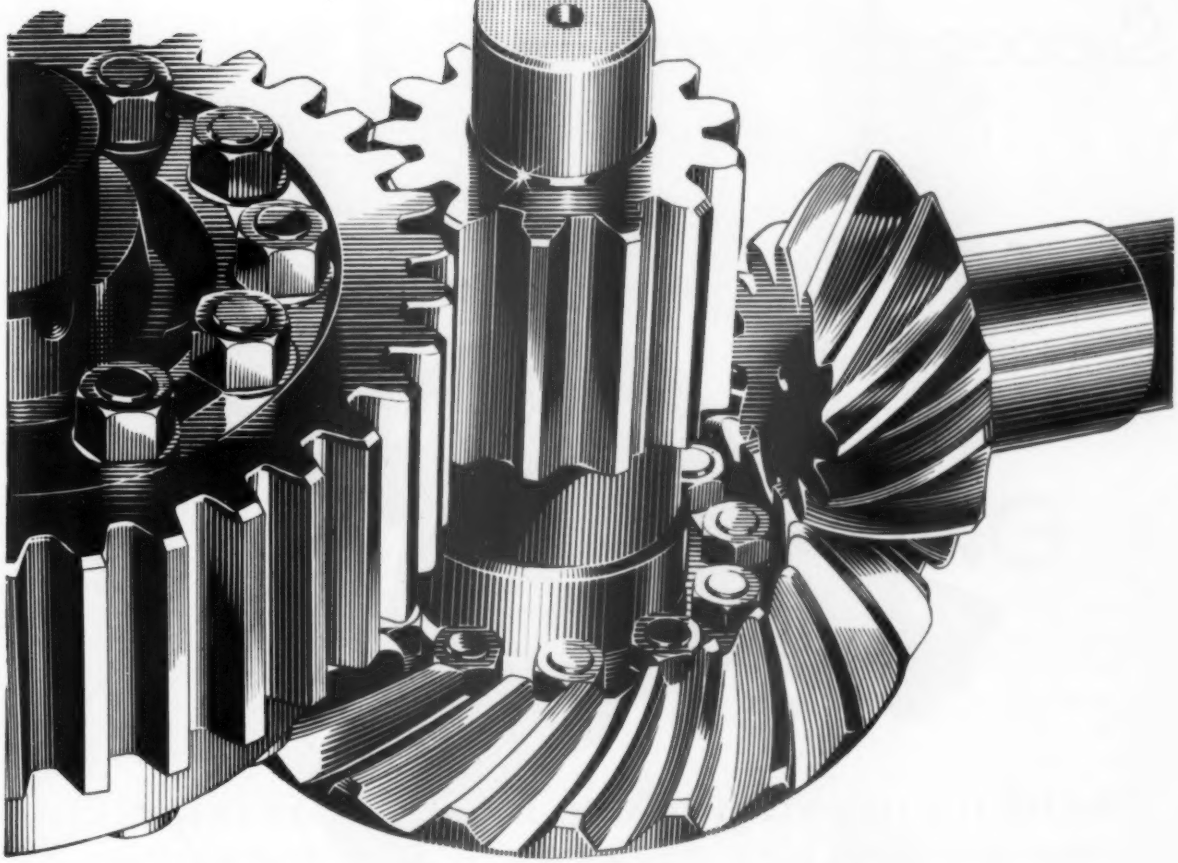

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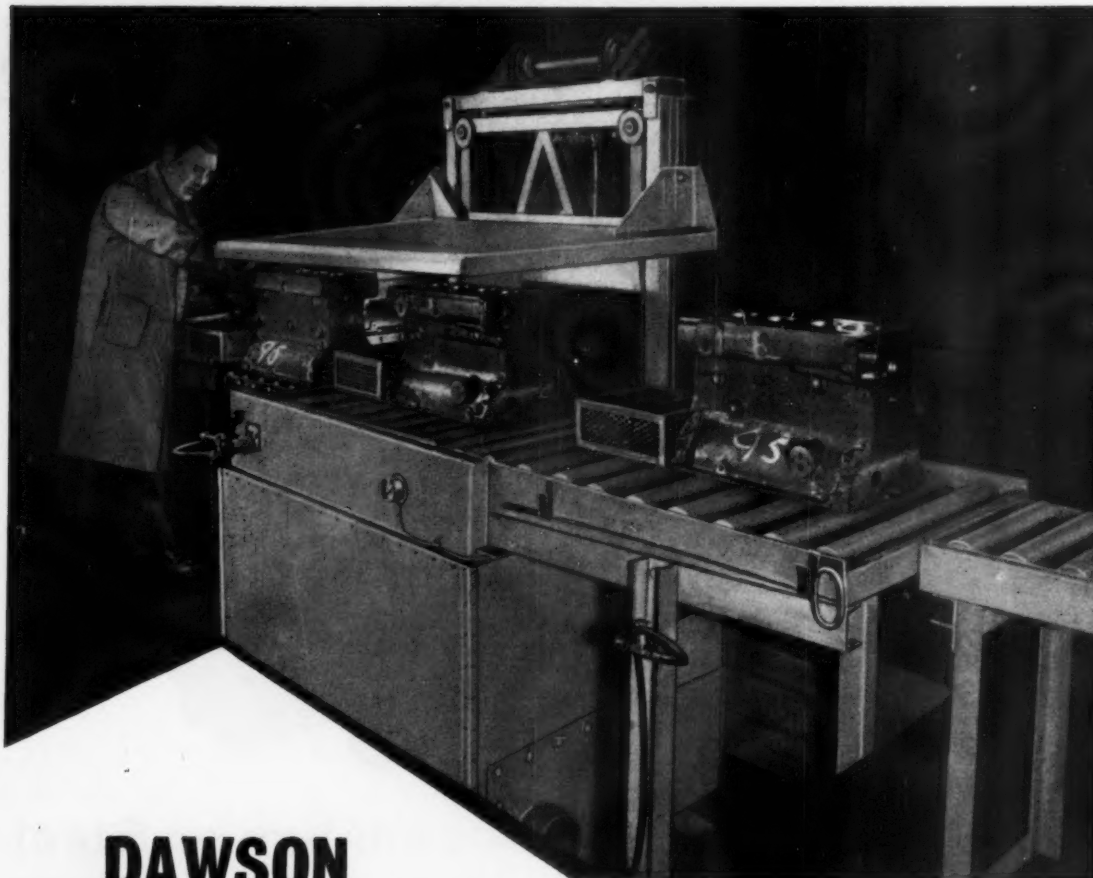

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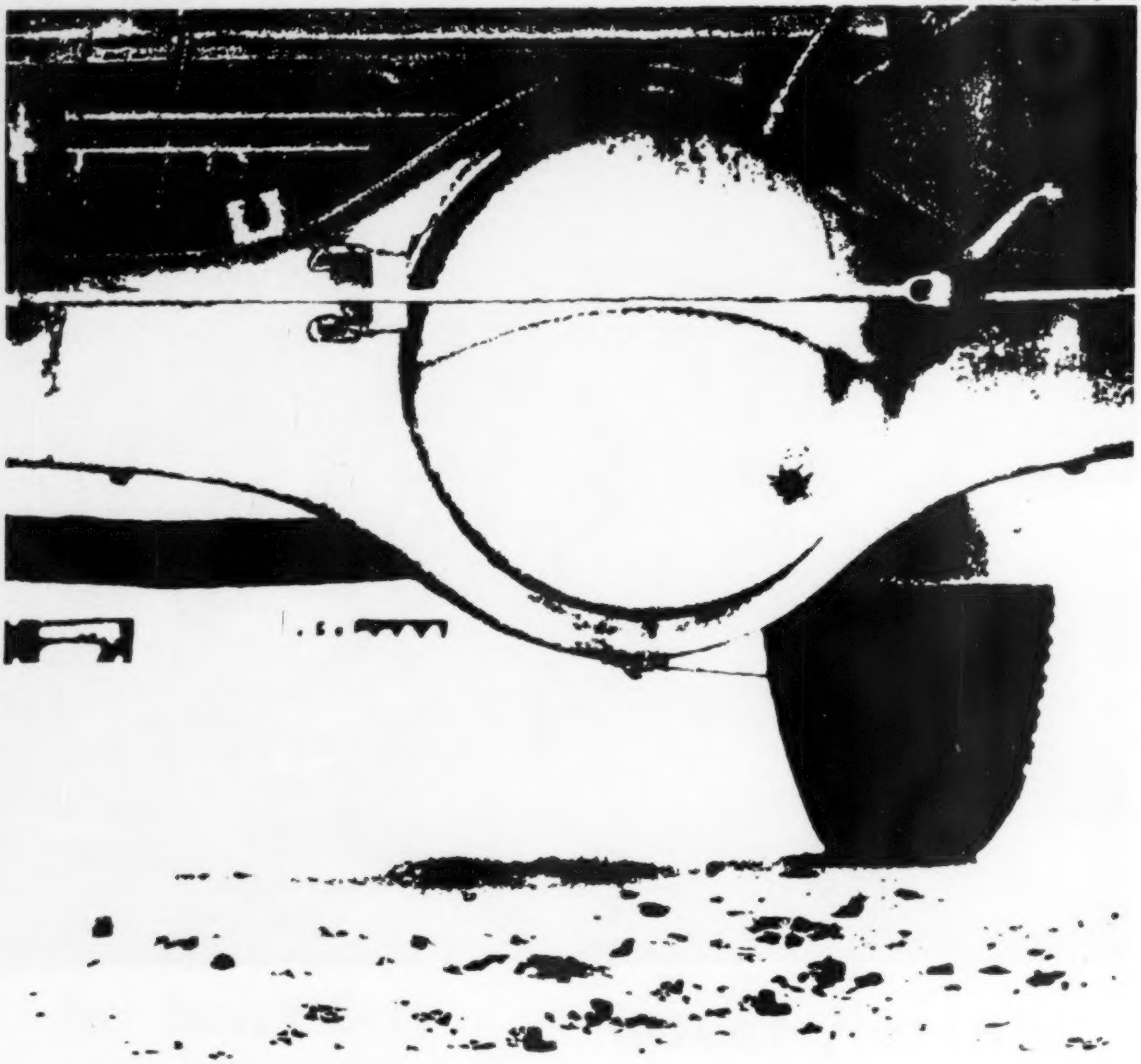

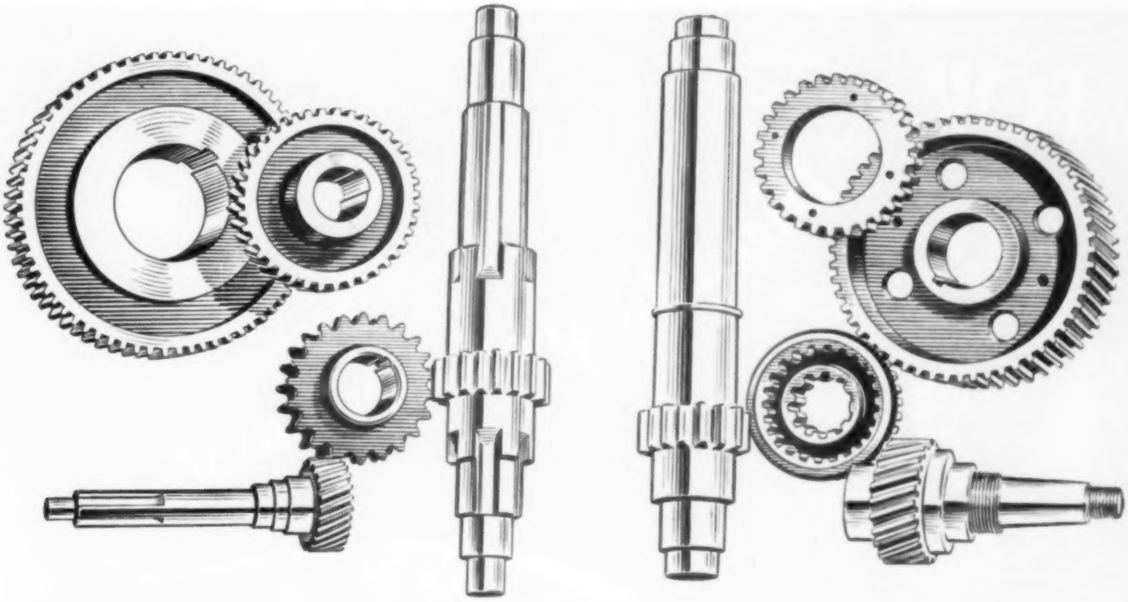

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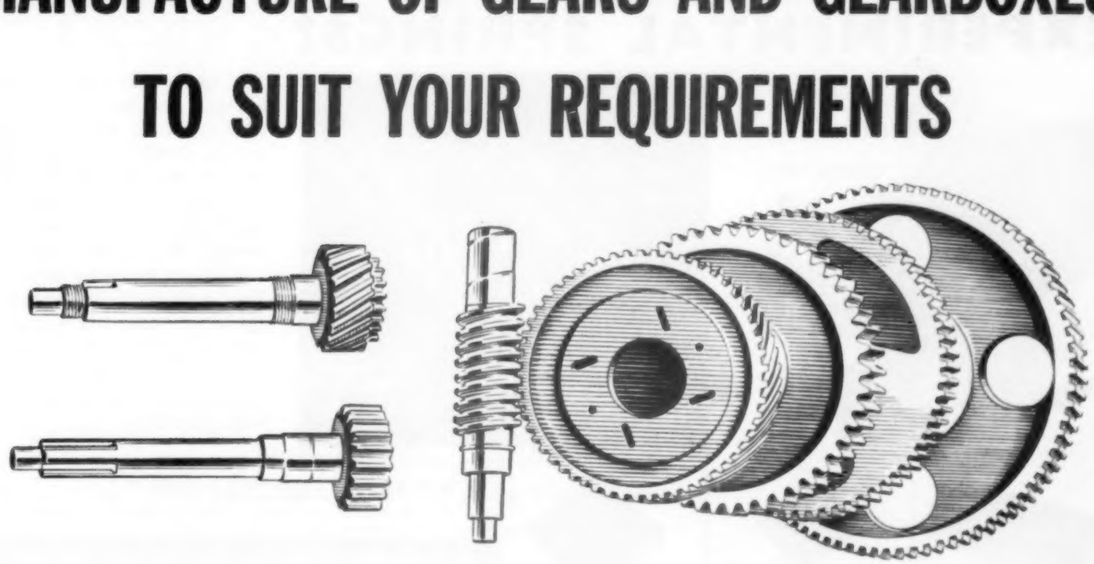

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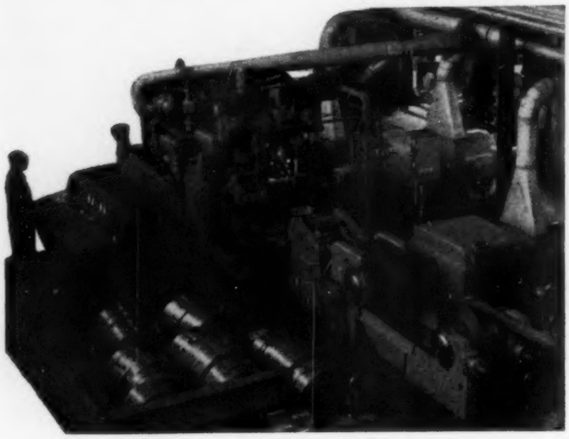

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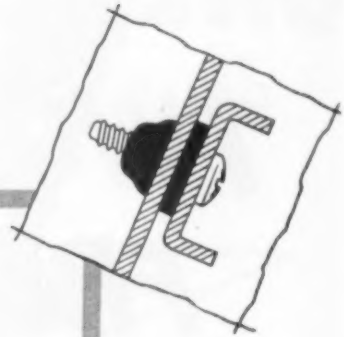

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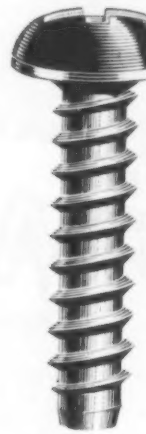

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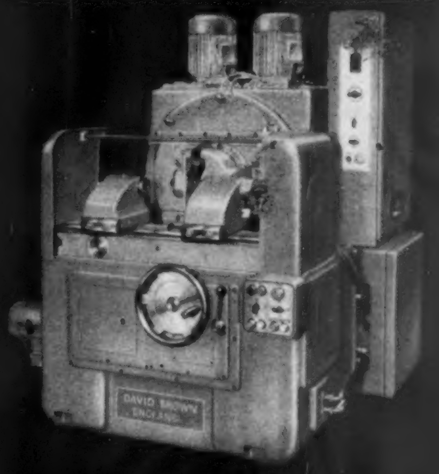

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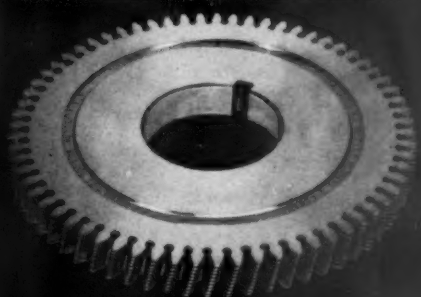

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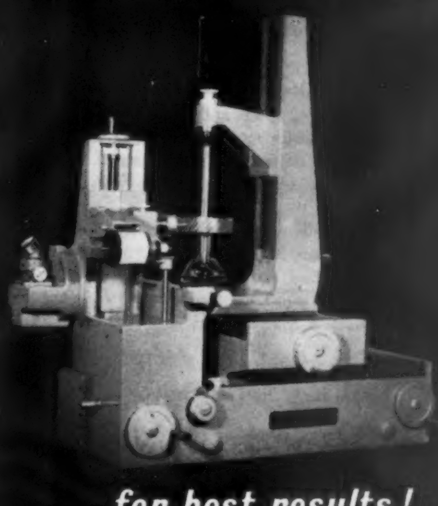

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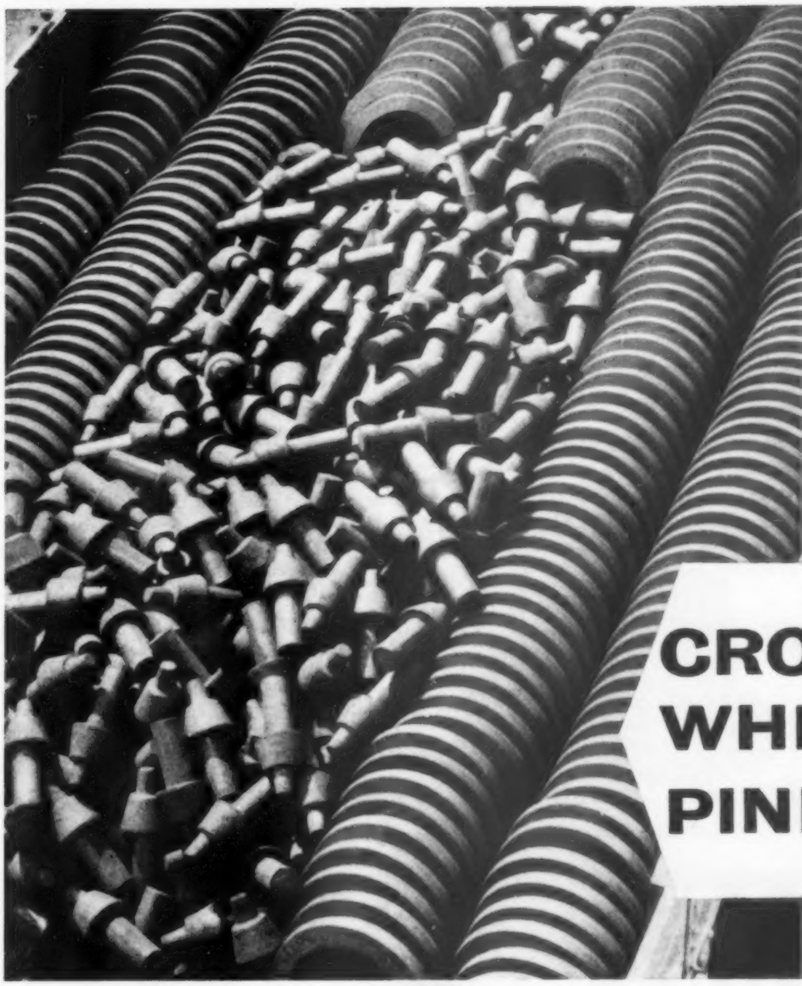

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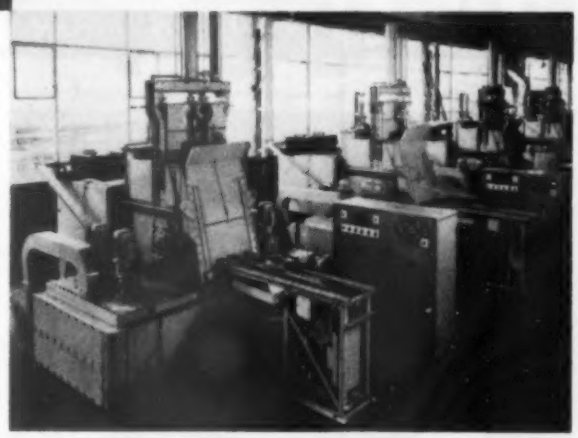

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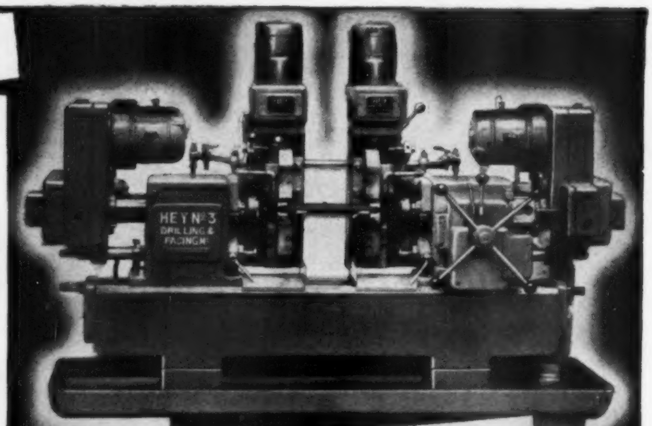

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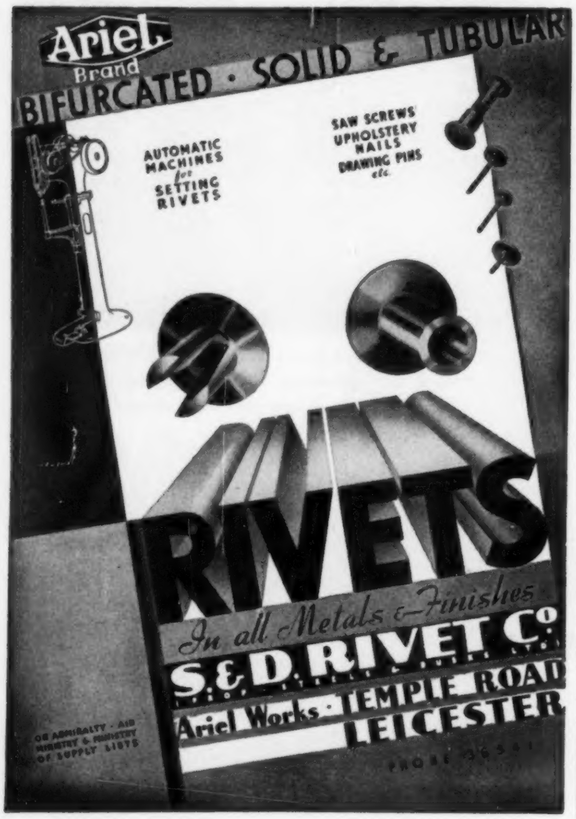

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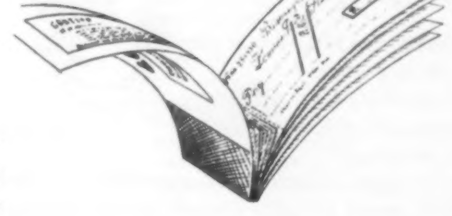

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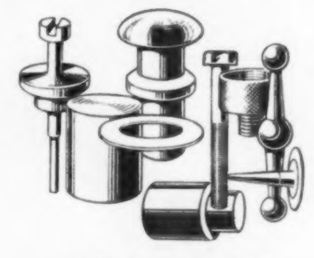

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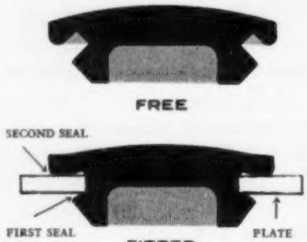

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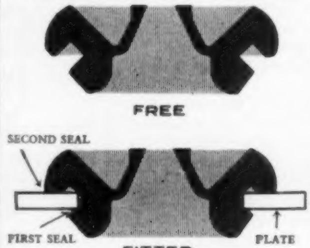

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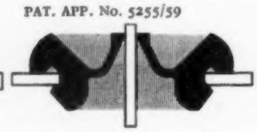
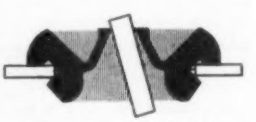


Large grommet in use, sealing a thin control rod



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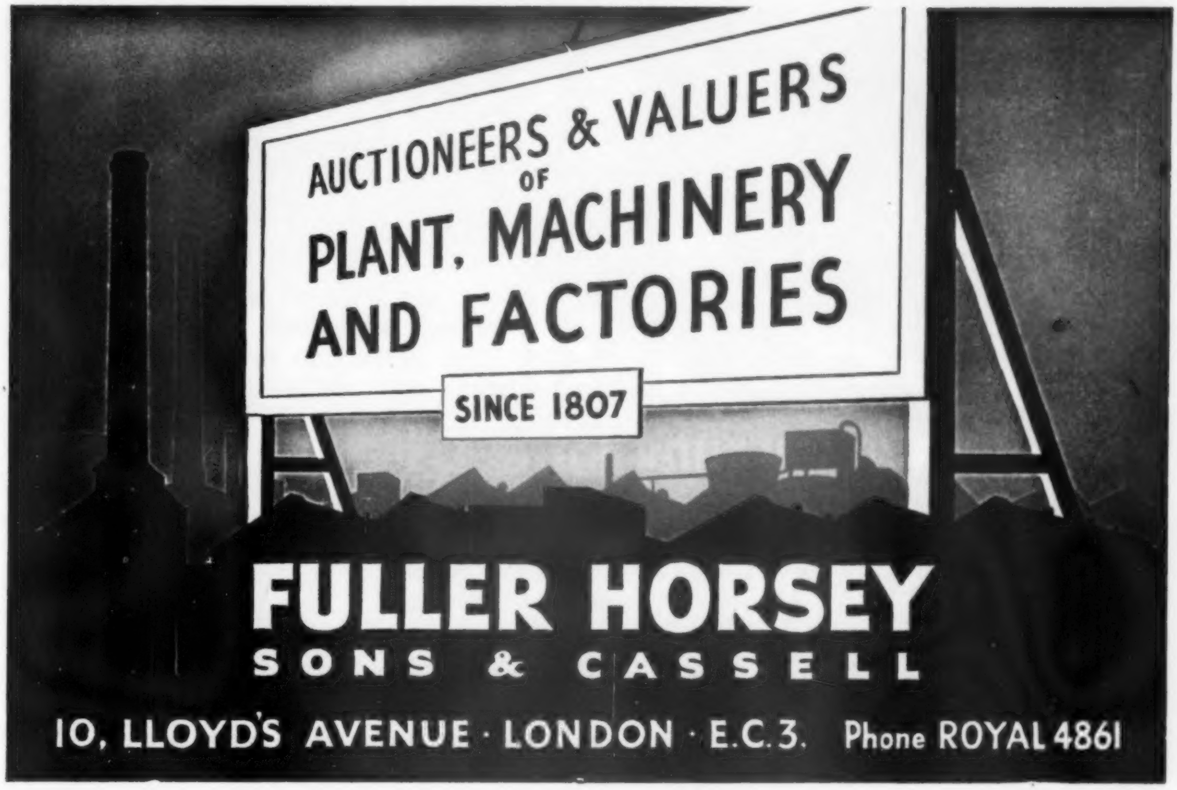

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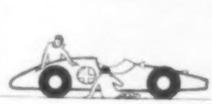

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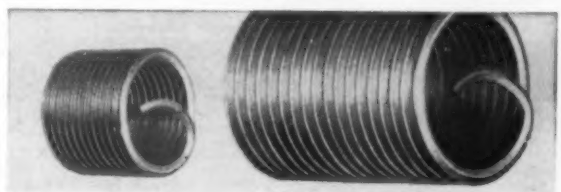

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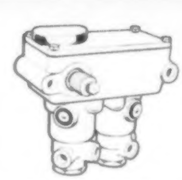

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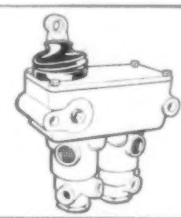

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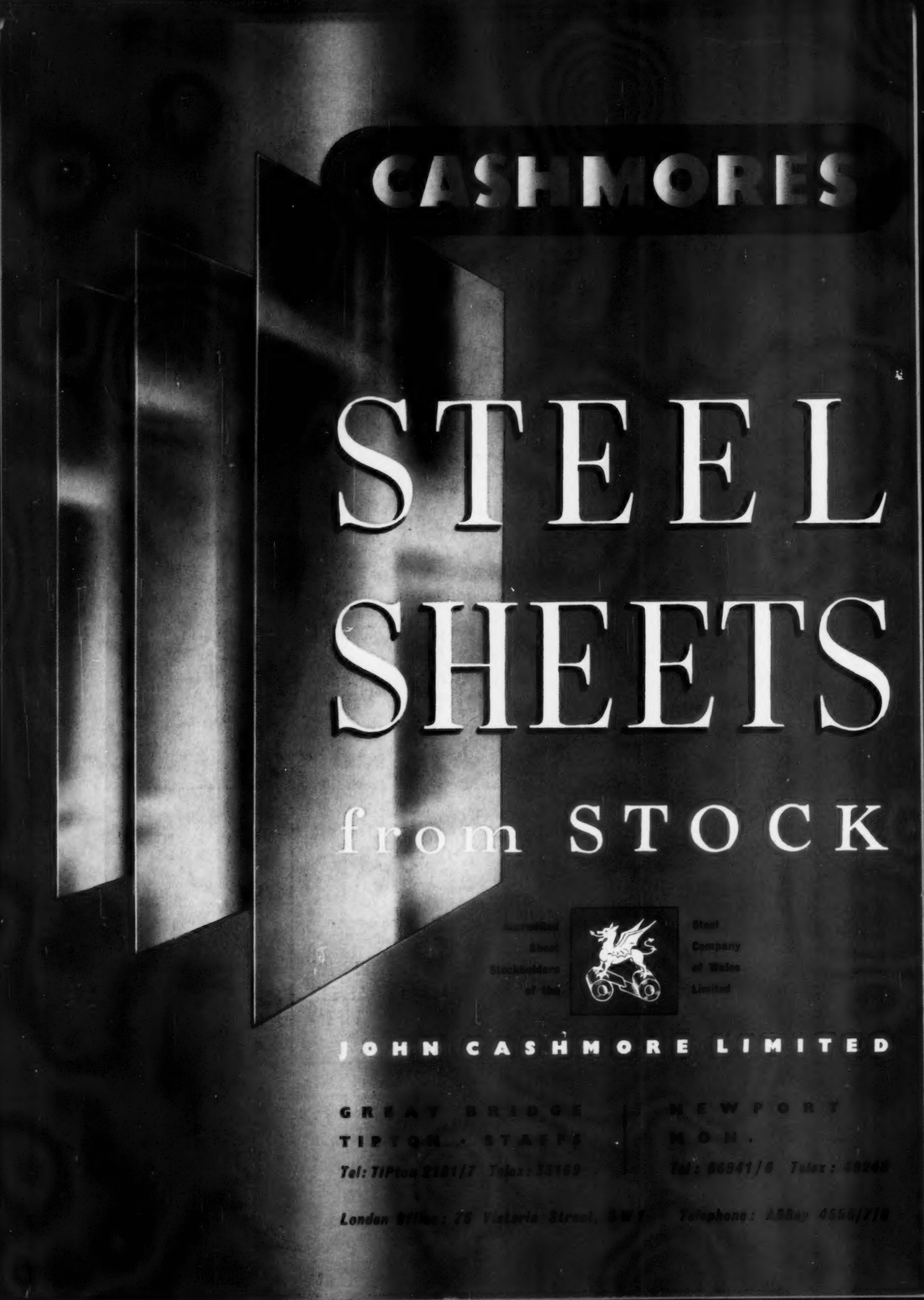
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